THE EFFECT OF ACOUSTIC VIBRATIONS
ON THE HEAT TRANSFER COEFFICIENT IN A CLOSED AIR CHAMBER

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SUMMARY

The purpose of this investigation was to determine whether the introduction of acoustic vibrations into a closed annulus of air produces a measurable increase in the magnitude of the heat transfer coefficient in the annulus.

The air chamber was defined by the walls of two concentric brass pipes, with the annulus blocked at both ends. The diameters of the two pipes were selected so that the effects of convection in the annulus were negligible. A cylindrical heater provided a constant flow of heat through the inner pipe wall. The input to the heater was measured, and, using temperatures measured with thermocouples installed in the pipe walls at the interface with the annulus, the values of the heat transfer coefficients at both walls of the air chamber were computed. These values were averaged and the resultant value was denoted h_{av}.

With conditions otherwise unchanged, vibrations produced by an audio oscillator and observed on the screen of an oscilloscope were then introduced into the bottom of the air chamber through a driver unit from a loudspeaker. The oscillator was adjusted so as to obtain the condition of resonance in the air chamber at the various frequencies selected for test. Using the temperatures measured under these conditions, an approximate correction was made for the energy introduced into the air chamber by the driver unit, and the heat transfer coefficient at each side of the annulus was computed again. The average of

the two values obtained was denoted h^*_{av}. A comparison of h_{av} and h^*_{av} for the tests at various frequencies indicated that the introduction of vibrations into the air chamber caused a maximum increase in h_{av} of about 12 per cent.

The experiments indicated that, at all frequencies tested, there is an effect on the heat transfer coefficient due to the introduction of vibrations produced by typical laboratory equipment. The maximum effect of the vibrations occurred at a frequency of about 2,500 cycles per second, with the magnitude of the effect decreasing at resonant frequencies either higher or lower than that value.

An increase in the frequency of the oscillations produces more nodes along the walls of the annulus, and consequently contributes to a larger increase in h_{av}. Typical loudspeakers, however, produce an accurate frequency response only in the middle range of acoustic frequencies. The intensity of the signal from the driver, thus, can be expected to rise rapidly during very low frequencies, to remain relatively constant during the middle frequency range, and to drop sharply at higher frequencies. The increasing effect of vibrations from zero to about 2,500 cycles per second appears to reflect a combination of the increase attributable to each of these factors. At frequencies higher than about 2,500 cycles per second, the effect of the higher frequencies in increasing h_{av} is offset by a decreasing intensity of the signal from the driver.

This experiment has established the existence of sufficient effect of acoustic vibrations on the heat transfer coefficient to warrant further research into the problem. If the equipment used in the experiment were refined somewhat, more conclusive evidence as to the magnitude of the effect produced could be obtained. A more exact analysis of the frequency response of the driver is suggested as a means of reducing uncertainty in the magnitude of the correction required to compensate for the energy input of the driver during the tests. The experiments should then be restricted to the optimum frequency range of the driver, and should provide some means for measuring the energy input of the driver concurrently with the conduction of tests.

CHAPTER I

INTRODUCTION

The history of the study of heat transfer by convection has been a continuous search to understand the behavior and characteristics of the heat transfer coefficient. Isaac Newton, in 1701, established the simple mathematical relationship which describes the flow of heat by convection, q, through a surface of area A, where t_1 is the surface temperature and t_2 the temperature of the bulk of the fluid, as

$$q = h\ A\ (t_1 - t_2),$$

thereby defining the heat transfer coefficient, h.

It was immediately apparent that determination of the value of h is exceedingly difficult. For some time the coefficient was believed to be a property of the flowing gas or liquid which was involved in convective heat transfer (1). In 1912, Langmuir introduced the concept of the existence of a thin laminar sublayer, or stagnant film, of fluid surrounding electrically heated wires, through which heat transfer occurs, not by free convection, but by conduction and radiation alone (2). Subsequent work in the field has, of course, enlarged upon this theory to establish that, where a wall separates two fluids (other than liquid metals) of different temperatures, the greater fraction of resistance to heat transfer occurs in an insulating laminar sublayer of fluid which invariably exists between the bulk of each fluid and the separating wall (3). The steep temperature gradient in this laminar layer and the thickness of the layer itself affect the magnitude of the heat transfer coefficient and, consequently, influence the rate of heat transfer by convection.

Especially during the past half a century, there has been a continuous and extensive effort exerted in attempts to determine the factors which account for the magnitude of the heat transfer coefficient and to establish dependable methods for predicting and controlling its behavior. During this period, it has been established that the thickness of the laminar layer is not definitely, nor easily, defined (4). There exists, for instance, no sharp boundary between the laminar film and the turbulent core of a stream of fluid passing through a pipe; the temperature gradient from the surface gradually tapers off to zero at the bulk of the fluid (5). The temperature gradient in the laminar layer is known to be steep, but the slope is not infinite (6). The thickness of the layer appears to vary with time, but assigning an average thickness to the laminar layer and to the buffer zone between the sublayer and the core of the fluid is feasible (7). Many of the factors which affect the magnitude and nature of the coefficient have been established. For instance, the thickness of the layer is known to decrease with increasing velocity. Similarly to the increase of the temperature gradient, the velocity gradient decreases steeply, but steadily, from the bulk of the fluid to zero at the wall surface. Many of the properties of the fluid are known to have a bearing on the value of the heat transfer coefficient, such as viscosity, density, thermal conductivity, specific heat, coefficient of expansion, and whether or not the fluid is changing phase (8, 9). The coefficient is likewise influenced by the nature of the confining surface -- dimensions of the conduit, shape factors, surface roughness, and thickness of the body in contact with the fluid. Any agitation of the fluid is also known to affect the coefficient (10).

The number of factors which must be considered in predicting the behavior of the heat transfer coefficient suggests the complexity of attempting to obtain a mathematical formulation of film coefficient behavior. In fact, solution of the differential equations which describe convective heat transfer has been possible only in a few very simple cases, and then only with the aid of simplifying assumptions (11). For this reason, recourse has been made to the principles of similarity and analogy. Dimensional analysis has provided useful dimensionless parameters which have afforded a degree of generality to the formulation of empirical relationships describing the various types of convection problems. Reynold's velocity and momentum analogies have accounted for much progress in convective heat transfer research; Prandtl, Taylor, Von Karman, and others have contributed important data through the impulse and boundary layer theories (12); Taylor first suggested the vorticity analogy; Bakhmeteff suggested analogy in terms of kinetic energy (13); the mass transfer analogy has been found appropriate.

Concurrent with and contributing to this same progress in the field has been a tremendous increase in the applications of convective heat transfer to industry. The effect of the insulating sublayer next to a wall confining a fluid, measured by the magnitude of the heat transfer coefficient, has become exceedingly important from two opposite viewpoints. In industries where heat exchangers are in wide-spread use, introduction of methods which will increase the heat transfer coefficient and provide more rapid and efficient transfer of heat has become economically desirable. Opposing this concept is the desire to reduce h in some applications in order to take advantage of the insulating effect of the boundary layer. This type of problem has become much more prevalent and more critical in the past two decades with the advent of metals which can withstand higher temperatures, permitting the introduction of high-temperature gas turbines, jet engines, and high-pressure steam power plants. Many instances have arisen where the insulation of the boundary layer is desirable, as the margin of safety in the design of equipment. If unknown or uncontrollable factors tend to increase the heat transfer coefficient, either consistently or in unpredictable fluctuations, the very feasibility of the equipment may be thereby imperiled. Control of the behavior of the heat transfer coefficient in these instances becomes even more important than where a higher h is desirable in the interests of efficiency.

Many different theories have been employed in attempts to increase the value of the film coefficient. Increasing the velocity of a fluid flowing in tubes or over plates has proved effective in this respect, but a relatively definite optimum exists, when the increase in pumping costs required to increase the velocity further can no longer be justified by the effect produced on the heat transfer coefficient. Varying the viscosity, density, specific heat, or other properties of the fluid may be effective, but is not possible in most cases because the nature of the installation will usually determine the properties and general temperature range of the fluids that are used.

The use of turbulence promoters in heat exchangers to accelerate the transfer of heat has received considerable attention. Nagaoka and Watanabe (14) have summarized the results of using coiled wires, spirals of various pitches, and propellor-shaped baffles in heat exchanger tubes to cause an increase in the heat transfer coefficient. Although such devices have proved practical for many applications, a continual balance must be made between the advantage gained in more rapid heat transfer and the disadvantage produced by an increased pressure drop caused by the turbulence promoter itself. Dunlop and Rushton (15), for instance, report on the use of baffles which increased the power absorbed in pumping by as much as five times. Colburn and King (16) conclude that the fluid-friction heat-transfer analogy is so definite that the heat transfer coefficient can be predicted if only the pressure drop caused by a certain baffle is known. Blow steam has been used in an attempt to scour off the laminar layer on a tube wall, thereby reducing the resistance to heat flow (17). Martinelli and Boelter (18) vibrated a heated cylinder mechanically in a tank of water; the temperature difference between the cylinder and the surrounding water was varied by between eight and 45 degrees F. by varying the magnitude and frequency of the vibrations. Arno Andreas (19) received a German patent in 1942 which recommends vibrating an entire heat exchanger, or its walls, at an amplitude of one to five millimeters and a frequency of at least 1,500 vibrations per minute. West and Taylor (20) report that h was increased by 60 to 70 per cent, with an accompanying maximum increase in power consumption of 30 per cent, by producing pulsations in the turbulent flow of water through the tubes of a heat exchanger. Tests were conducted while the Reynolds number of the flow varied between 30,000 and 85,000. The cold water passed through a reciprocating pump into a vertical cylindrical tank, in which a chamber containing compressed air could be maintained over the level of the water. The air pressure in this chamber was varied so as to obtain the desired amount of damping of the pulsations in the water. These investigators credit the pulsations with reducing the thickness of the laminar layer by increasing the overall flow rate and by the radial diversion of part of the kinetic energy during pulsations.

West and Taylor also summarized concisely the practical considerations involved in the use of such a system or of any type of turbulence promoter in increasing the heat transfer coefficient. They point out that the advantage gained will be reflected in a smaller area of the heat exchanger with consequent lower initial cost. To be weighed against this advantage are the increased investment occasioned by initial installation and subsequent maintenance of the turbulence promoter itself, the possible reduction in the life of the equipment brought about by using the turbulence promoter, and the net change in power consumption for the same heat performance.

Considerable work has been done in connection with fluidized beds in attempting to increase the heat transfer coefficient. Dow and Jakob (21) have reported on an experiment involving the transfer of heat to a fluidized bed wherein the predicted value of h was exceeded by a multiple of several times. Dow and Jakob theorize that the large value of h encountered in experiments with fluidized beds is caused by the bombardment and subsequent derangement of the laminar sublayer by the solid particles suspended in the stream.

The possibility that acoustic vibrations can be used profitably to control the value of the heat transfer coefficient appears very promising. As with mechanical vibrations, this effect has significance either as a means of increasing the rate of heat transfer for more efficient exchange of heat or as a means of controlling the rate of heat transfer by eliminating unpredictable vibrations which may be causing excessive or unpredictable values of h.

The concept of the effect of a controlled standing wave on the transfer of heat from a solid body was the subject for a series of interesting experiments conducted by Kubanski (22, 23). He reported upon several tests in which he directed a sound wave of variable frequency at a section of one-inch pipe which was suspended in the sound wave. The pipe was provided with a heating coil and a means of temperature measurement. A photographic study was made of the characteristics of the transfer of heat from the pipe when the conditions of resonance prevailed. Kubanski reports measuring a rise in the value of h by "twice or more" during some of these tests. He claims to prove with his experiments the theory of "acoustic wind" or "acoustic velocity," and thereby to explain the reason for the increased rate of heat transfer under the conditions described.

According to Eckart (24), "acoustic wind" was reported by Lord Rayleigh many years ago, when he observed that it was possible to hold a tuning fork to a resonator, and to direct the air currents that were produced when the tuning fork was sounded in such a manner that a candle could be blown out. Several other investigators have observed the phenomenon of "acoustic wind" and have verified its behavior (for instance, 25, 26).

Kubanski explains that when a sound wave is at resonance with a body, there is a movement of air from the antinode to the node. When the pipe was suspended so that its axis was parallel to the axis of the wave, a small scrap of paper placed on the pipe at an antinode was physically displaced to either side of the antinode, along the axis of the sound wave, by this current of air. At each node, two streams from the antinodes at either side meet, are deflected perpendicularly away from the pipe in a single stream, and thus create the "acoustic wind" phenomenon. Further evidence of the nature of the phenomenon is apparent in the observations that temperatures measured at the nodes are higher than at the antinodes. The effect of this temperature difference, in fact, made the photographic study possible, since the increased temperature at the nodes lowered the density of the air at those points relative to the density at the antinodes, and produced dark areas on the photographs.

Kubanski predicts that the flow away from the pipe is probably in the form of concentric rings. The warm air next to the heated pipe which is moved along in the boundary layer away from the antinode is replaced by relatively cold air from the surroundings, which causes the increase in the rate of heat transfer. If the vibrations are of sufficient magnitude, the exchange of air becomes so rapid that turbulence is established in the boundary layer. Experiments have indicated previous to this time that a relation exists between the velocity of the air in the "acoustic wind" and the frequency of the vibration, but two distinctly different proportionality factors apply to that relationship, depending upon whether the movement is laminar or turbulent in the boundary zone (25). Much of the concept presented by Kubanski was predicted by Anrade (26). Anrade used smoke particles to indicate the path of the air currents.

Kubanski reports that the effect on the heat transfer coefficient depends upon the intensity and the frequency of the signal, plus the magnitude of any convective currents which exist. He found that, as the frequency increases, past a certain optimum, the power output of the vibrator used was reduced sharply, so that those two effects were not cumulative at the higher frequencies. For this reason, Kubanski's experiments were limited to a relatively narrow frequency range.

Earle (27), in discussing the implications of the increased use of ultrasonics in industry, describes experiments being conducted in the vibratory firing of pulverized coal. A strong beam of sound waves is directed into a column of burning coal particles. Apparently the vibrations, in stripping away the laminar layer around the burning particles, result in more complete oxidation by exposing the particles to additional oxygen. The phenomenon discussed by Kubanski should also result in more rapid transfer of heat to the surroundings, so that more of the heat of combustion can be liberated before the products of combustion enter the stack.

The phenomenon of sudden deterioration of the tail pipes of after-burners in jet aircraft is a fairly well-known occurrence. It is most convincingly explained as caused by a disturbance of the laminar layer in the pipe by the acoustic vibrations carried in the high noise level, or "screech," emitted through the afterburner. Since there is every indication that any such effect in raising the heat transfer coefficient would reach a maximum when the screech attained a condition of resonance with the afterburner, it is plausible that the presence or absence of conditions for resonance may explain the unpredictable occurrence of this phenomenon.

The same phenomenon has been suggested as the cause of the over-heating of an internal combustion engine which accompanies detonation. If the vibration induced by detonation is of a proper nature to displace the laminar layer, there would result a more rapid transfer of heat into the coolant and a consequent "over-heating" of the engine.

From the many possibilities suggested in the foregoing summary, the effect of acoustic vibrations on the heat transfer coefficient at the boundaries of a column of air has been selected as the subject for study in this paper for several reasons. The use of acoustic vibrations as a positive means for obtaining more rapid heat transfer is proposed as a possibility which could become an exceedingly valuable expedient. Acoustic vibrations can be generated at comparatively small expense and with compact, light-weight, and portable equipment. Such vibrations could be utilized to effect improved heat transfer without materially reducing the life of equipment in the usual type of heat exchanger. If effective, such a means of expediting heat transfer might profitably be applied, for instance, to applications such as radiators in water-cooled Army tanks, where space and weight of the cooling system are critical factors, and exposure to the exterior of the vehicle of a beehive, or core-type, radiator is undesirable. If such a means of affecting rapid heat transfer is practical, it could, indeed, be applied to almost any existing heat transfer equipment with very little modification of existent systems.

With these factors in mind, it was concluded that determination of the effect of sonic vibrations on the heat transfer coefficient presents an important problem. Apparently there has been little or no investigation into the possibility of using simple equipment, such as might be readily available in most laboratories, to produce acoustic vibrations and to measure the effect of these vibrations on the rate of heat transfer. Kubanski's experiments were reportedly carried out meticulously, with carefully calibrated apparatus and in a narrow frequency range; a survey of current literature did not reveal any other experiments involving the specific problem of acoustic vibrations and their effect on the heat transfer coefficient.

The decision was made, therefore, to conduct this type of investigation. The objective was to be the determination of the magnitude of the effect on the heat transfer coefficient of acoustic vibrations produced with ordinary laboratory equipment. Every effort was to be made to keep the entire system as simple as possible, avoiding complicated or especially constructed equipment and apparatus which could not be readily duplicated in a practical application. The conduct of such an experiment should indicate whether the effect of such vibrations is substantial enough to be significant without careful calibration and control of all components of the apparatus involved; whether sonic vibrations at ordinary intensities and frequencies produce a consistent or a material effect on the magnitude of the heat transfer coefficient, or if this effect is transitory and applicable only to a meticulously regulated system. Such an experiment should also indicate which of the variable factors in the apparatus are the most critical in determining the magnitude of the effect, so that perhaps any future experiments can refine the procedure and the apparatus in such a manner as to obtain more consistent or emphatic results.

In conducting an experiment of this type, isolation of the effect of acoustic vibrations in changing the value of h appeared desirable. Either free or forced convection accompanied the acoustic vibrations in all of Kubanski's experiments. Other examples which have been mentioned of efforts to increase the value of h have always involved other factors, such as physical movement of the heated surface or velocity influences. Uncertainties exist, however, in present-day concepts regarding the behavior of the transfer coefficient, and data concerning its behavior are empirical. Therefore, in attempting to analyze the effect of acoustic vibrations on the coefficient, the desirability is indicated of eliminating not only all the unknowns over which control is possible, but also all of the factors which are known to affect the behavior of the coefficient, lest the magnitude and behavior of any effect due to acoustic vibrations be confused with the effect caused by some other factor about which knowledge may be incomplete. For this reason, the equipment was designed to eliminate to as great an extent as possible, all other controllable effects, such as convection and velocity.

CHAPTER II

EQUIPMENT AND INSTRUMENTATION

The frequencies, temperatures, amplitudes, materials, and equipment which were selected as the basis for constructing the apparatus are believed to be those such as might be readily available for a practical application. The decision was made to utilize a closed column of air as the medium to be vibrated, since the analysis of vibrations produced, the attainment of a standing wave, and the measurement of the heat transfer involved appeared simpler and more certain with a closed system than with one or both ends of the chamber open (28). Also, there will be no diminution of the sound wave in this type of system, since the usual distribution of the wave over a wider area after the wave is propagated is not possible.

A cylindrical arrangement appeared most practical for the test section. A heated inner cylinder, surrounded by a thin annular ring of air, enclosed by an outer pipe, appeared to provide the optimum arrangement with respect to end effects and heat leakage. The concentric cylinders were mounted vertically to establish axial temperature symmetry, in the event that any convective currents are created in the air chamber. In tests conducted with a similar set of concentric tubes arranged horizontally, Beckmann (29) found it necessary to arrange a cylindrical heater slightly eccentrically in order to obtain uniform temperature.

Both ends of the annular arrangement of tubes were blocked with a plastic plug of low conductivity. The surface which was to confine the air column on the upper plug was finished as carefully as possible so that the sound waves would be reflected perpendicularly from that surface. The leads to the heater and to the thermocouples were taken through one end plug; the other was threaded to receive a permanent-magnet driver unit from a loud-speaker, as a means of introducing the acoustic signal into the apparatus. A water jacket was placed concentrically around the outer tube. Figure 1, in Appendix B, shows an assembly drawing of the test section; Figure 2 is a photograph of the apparatus, disassembled after the tests were completed.

Any movement of the air in the annulus, except for such movement as may be caused by the vibrational effect itself, is undesirable and should be avoided. To accomplish this purpose, it was necessary to assure that no natural convective currents exist, so that heat transfer through the air layer will be by conduction alone.

Determination of the relative size of the tubes depended upon this requirement that convective currents be minimized. In this respect, the experiments of Beckmann (29), mentioned above, were found to be helpful. Beckmann studied the extent of convection between coaxial cylindrical tubes, using an apparatus which is very similar to the one employed in this experiment. By changing the diameter of the outer tube, he recorded data at seven different diameter ratios. He established various values for k_c/k, where k_c indicates the proportionality factor for heat transfer through an annular layer of gas by conduction and convection, and k the value for transfer by conduction alone[1]. Beckmann computed values of the Grashof number with D_1 as the characteristic length (i.e., the diameter of the inner tube):

$$Gr_1 = \frac{\beta g}{\nu^2} D_1^3 (t_1 - t_2).$$

Using temperature differences of from 10 to 25°C., Beckmann established data for plotting ln (k_c/k) versus ln $(Gr)_1$ at various diameter ratios. Such a chart is presented in Beckmann's paper, and is also reproduced in Jakob's review of that paper (30).

For the experiment presently considered, determination by Beckmann's method of the tube sizes to be used involved computing Gr_1 for the diameter ratios formed by various pairs of conveniently available pipes and tubes, and assuring that k_c/k for the combination chosen be not greater than unity. In this manner, two pipe sizes were decided upon for which Gr_1 is 51,500. According to Beckmann's chart, k_c/k is equal to unity for the diameter ratio chosen at Grashof numbers up to about 80,000 (30), so that convection is assumed to be negligible in the test section. Computations are indicated in Appendix D; a summary of the physical characteristics of the test section is presented in Appendix E, which lists pertinent specifications for the pipes.

This analysis was checked by the method suggested by Jakob (31), using a mean area and the procedure for determining the magnitude of convection in vertical cylinders, as adapted by Jakob from Mull and Reiher. In this case, also, k_c/k was computed as equal to unity. This procedure appears to be equivalent to correcting the cylindrical

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[1]Symbols used in this thesis are defined in Appendix A.

dimensions so as to consider the air layer as a developed plane. If this is acually done, with L, the thickness of the air layer, as the characteristic length, Gr_L is 531.2. For such cases, convection is considered negligible when Gr_L is less than 2,000 (32). Another more general check is indicated in considering the actual 0.222 in. thickness of the air layer in light of Jakob's statement that "it is obvious that convection will increase with $D_2 - D_1$; it becomes negligible when $D_2 - D_1 < 1/4$ in., because of the effect of air viscosity." (33)

In the computation of the Grashof number, the value for β is taken as $1/T$, where T is the absolute temperature of the bulk of the gas. This relation is assumed to be true for perfect gases at temperatures near room temperatures (34). In the hope of determining whether the effect of acoustic vibrations has practical application in air at moderate temperatures, a decision was made to restrict the maximum temperature in the test section to 150°F., and to use a value for β of $1/T$.

The relatively low temperatures decided upon permitted the use of Plexiglas as a material for the end plugs. This material is readily machined. It possesses thermal conductivity of about 0.14 B/hr-ft^2-°F.

The heating element used was a commercial cartridge-type heater. It consisted of a helically-grooved ceramic core, wound with nichrome resistor wire. The element was surrounded by densely packed refractory material and centered in a thin brass sheath.

Several considerations dictated enclosing the heating element in a concentric brass pipe which was very slightly larger in diameter (0.072 in.) than the sheath of the heater. Since measurement of the temperature of the pipe was desired at the outside surface of the heated tube, the thermocouple junctions were conveniently installed from the inner side of this pipe through a very small hole drilled through the pipe. The junction was soldered to the pipe where it protruded from the hole, and the place of attachment was then polished smooth. This procedure assured taking temperature measurements at the surface of the pipe, but left undisturbed the surface of the pipe which was to be exposed to the air annulus. Four thermocouples were mounted at equal intervals, radially and axially, along the heated portion of the tube. The insulated leads were then led between the heater and the pipe wall to the Plexiglas end plug and into the atmosphere through very small holes. In this manner, conduction along the leads was minimized, since the leads passed through a zone which was substantially isothermal for several inches before leaving the apparatus. Shims were used to center the heater in the pipe. Introduction of the thin annulus of air between this pipe and the heater introduced a temperature drop of about 10°F., but this temperature drop was inconsequential since the temperature measurement was taken at the outer surface.

Mounting the heater in the brass pipe increased the heating area considerably and provided a more favorable diameter ratio for determination of the effect of convection by Beckmann's method. Also, Jakob, in commenting on the use of cylindrical heaters during tests to determine thermal conductivity, advises: "It is even recommended to cover the heating coil by another metal tube which equalizes the temperature over the heater surface." (35) Compensation for the end loss effect of this heater-cover was accomplished by averaging the four equally spaced thermocouples to determine the wall temperature.

The driver end of the heating pipe was fitted with a hemispherical brass plug. A 1/4 in. thick wafer of asbestos insulation was placed between this brass plug and the end of the heater, in order to contribute more reliability to the assumption that heat flow from the heater is radial. The hemispherical plug was used in order to reduce the direct reflection of the signal into the cone of the driver, and thus to provide more flexibility in locating points of resonance.

Four thermocouples were also mounted in the outer brass pipe (Tube B) at intervals approximately corresponding to the location of the thermocouples on the inner pipe (Tube A). A small hole (number 55 drill) was bored from the outside of Tube B to approximately 0.012 in. from the inner surface. A groove 0.06 in. square was milled from each hole to the end of the pipe opposite the driver. A small piece of solder was melted in each hole, using an oxy-acetylene torch, and the thermocouple junction was heated and inserted into this molten solder. The insulated leads were then cemented into the grooves, and were led to the atmosphere through small holes in the end plug. A junction between the couples and the pipe metal was achieved in this manner at a point which was very close to the inner surface of the pipe, although in each instance the inner pipe surface was left undisturbed. As in the case of the thermocouples on the inner pipe, conduction along the thermocouple leads was minimized by bringing the leads through an isothermal area before they were led into the atmosphere. The two pipe surfaces in contact with the air annulus were polished to minimize the effects of radiation.

The water jacket was formed from a section of 2 1/2 in. black steel pipe, provided with fittings for entry and exit of the water. Two standard 2 1/2 in. caps were bored through the center to receive the outer brass pipe. One of these caps was soldered to that pipe near one end; the junction between the second cap and the pipe was sealed by means of a 3/16 in. "O" ring, and a metal slip ring was brought to bear on the "O" ring by means of bolts on four 1/4 in. studs which had been brazed to the outside of the cap.

The entire apparatus was placed in a large cardboard tube, 9 1/2 in. in diameter, and was surrounded with a minimum of 2 in. of expanded vermiculite insulation. A summary of the physical characteristics of the components of the test section appears in Appendix E.

A small motor-generator set provided the output to the heater. The output of the generator was about 18 volts; the output to the heater was controlled by introducing a voltmeter, a milliammeter, and a variable resistor into the motor-generator heater circuit. This arrangement permitted the use of a relatively high voltage to warm the equipment to the approximate temperatures desired quickly, after which a preselected voltage, based on an estimate of the heat flow required to maintain a desired temperature difference between the walls, could be established and maintained during the remainder of the test. The use of direct current was chosen in preference to alternating current to avoid any possible effect of a power factor, inductance, or eddy currents created in the nichrome wiring by the buildup and collapse of the alternating current field.

Vibrations were supplied to the driver, mounted in the bottom of the test section, by means of an audio oscillator. A cathode ray oscilloscope was connected across the line to the driver to permit observation of the signal between the driver and the oscillator, so that points of resonance could be identified and maintained during testing. A schematic diagram of the oscillator circuit is shown in Figure 3.

The temperature measurement system was based upon a desire to attain accuracy of about $0.1^{\circ}F$. In addition to the eight wall thermocouples, whose installation has been described, one thermocouple, mounted in a packing gland, was screwed into the side of the container in which the apparatus was located or into the inlet water line, depending upon whether cooling water or the ambient temperature was to be considered as the environmental temperature during a particular test. Junctions of all thermocouples were soldered, and the same spool of Leeds and Northrup 30-gauge iron-constantan wire was used throughout the apparatus. The method used in calibrating the thermocouples is discussed in Appendix F.

A Leeds and Northrup Type K-2 potentiometer was used, in conjunction with a Leeds and Northrup 2500 Type R reflecting galvanometer and an Epply standard cell. The galvanometer was mounted on a stand with a concrete-weighted base at a distance of approximately one meter from the scale. Considerably less effect upon the galvanometer of externally caused disturbances of various types was noted when the weighted stand was placed in a shallow box containing several inches of fine sand. Operating potential for the potentiometer was taken from dry cell batteries.

An iron-constantan thermocouple, maintained at the temperature of melting ice in a thermos bottle, was used as a reference junction. The thermocouples and reference junction were connected to the potentiometer through an industrial type thermocouple switch box. A schematic diagram of the thermocouple wiring circuit is presented in Figure 4.

Figure 5 indicates the arrangement of the equipment as it was installed for testing.

CHAPTER III

PROCEDURE

Determination of Procedure to be Used.--Experimental determination of the effect of acoustic vibrations on the heat transfer coefficient required, basically, the measurement of the heat flow through the tube walls and measurement of the wall temperatures. Measurements are necessary during a period in which there is heat flow but no vibration of the air column, and during a period in which the air in the column is being vibrated while the same heat flow is being maintained. Several possible methods of determining these data were considered, initially. Various possibilities were then discarded, during design of the apparatus and during testing, until the final test was evolved.

The first possibility rejected was measurement of the change of temperature of the water which passed through the water jacket, in order to determine the heat which flowed through the outer surface of Tube B. In discussing the design of the pipe walls in Chapter II, mention was made of the decision to limit the maximum temperature within the tube area to about 150°F. This temperature limit restricts the possible heat input by the heater. Assuming that q, the heat flow through the air column, is by conduction alone,

$$q = k_m A_m \frac{t_1 - t_3}{x_3 - x_1},$$

where k_m is the thermal conductivity of the air column at the mean temperature of the air,

$$A_m = \frac{A_3 - A_1}{\ln A_3 - \ln A_1} \quad \text{(the logarithmic mean area),}$$

t_1 and t_3 represent the wall temperatures at the inner and outer surfaces of the air annulus, and $x_3 - x_1 = 0.222/12$, the thickness of the air layer (36). Using this formula, an estimate of 15 B/hr. was computed as the maximum possible q for a 40° F. temperature difference across the walls, which is in excess of the highest temperature difference encountered during testing. Appendix E lists data relative to the water jacket which indicates that 15 B/hr. requires a flow rate of about 0.09 gallons per minute before a $1/3^\circ$ F. rise in the temperature of the water occurs in passing through the apparatus. Thus, the expected heat flow was obviously inadequate to permit maintaining any measurable flow rate in the water jacket which would provide, at the same time, a measurable temperature change of the water as it passed through the jacket. For these reasons, the possibility of measurement of the change in temperature of the water was discarded.

A second possible procedure was to maintain a flow of water through the water jacket which would be adequate to assure that the temperature at the outer surface of Tube B was significantly constant (presupposing that the water temperature is constant.) When this procedure is used, a sufficient flow of water is maintained to assure that all heat flowing through the wall is absorbed by the water without any appreciable change in the temperature of the water as it passes through the jacket. The heat flow at any point radially from the heater can be estimated by subtracting from the heater output any heat storage between the heater and the point under consideration. This method was used in Runs 11, 12, and 13. The only source of water available, however, was from a tap considerably removed from the city water main. The temperature of this water varied more than 7°F. during the day. In addition to the trend toward higher temperature during the day, there were slight irregular fluctuations of the water temperature. The results of the three tests tend to substantiate the results obtained by the method decided upon, but they are believed to be less conclusive. The system of computations used, because of its dependence upon a constant rate of change of wall temperature, appears to be very sensitive to fluctuations in wall temperature, and, in this instance, to fluctuations in the temperature of the water.

The actual test procedure decided upon involved thoroughly insulating the entire test section by placing it in a large container, surrounded with a thick layer of insulation. In this manner, the thermal resistance between the outer surface of Tube B and the environment was made quite large, and a time lag estimated at about six hours was introduced. In the absence of temperature control of the room where the experiment was conducted, the environmental temperature varied continuously through a daily range of about 25°F., so that it was never possible, of course, to obtain steady state conditions within the apparatus. However, the effect of environmental temperature was greatly reduced by the effect of the insulation. The rate of temperature change at the outside tube wall was found to be small, predictable, and comparatively steady during the time required for a test, with less fluctuations than would be experienced with less insulation, and with a more consistent rate of change than was obtained by introducing the temperature of the water outside of Tube B. The method which was used to compute the heat transfer coefficient is described below.

Conduct of Tests.--Temperatures of the tube walls were measured under three separate sets of conditions. These three sets of conditions will be called Case A, Case B, and Case C, for convenience in identifying the data recorded. Figure 6 indicates schematically the conditions which were assumed to prevail in each of these three cases, together with the basic formulas which were used in computing the data in the Appendix C, as explained below. Case A represents the flow of heat from the heater through the air column and tube walls when there is no signal being transmitted and, hence, no heat being generated by the driver. Case B indicates the conditions which prevail when there is no input to the heater, but there is a signal being emitted by the driver, with the consequent transmission of an amount of energy S (in B/hr.) through the air column to the tube walls. When there is a current flowing through the heater at the same time that the driver is transmitting energy into the system, the conditions of Case C prevail.

The flow of heat through the tube walls was assumed to be radial. Because of the magnitude of the thermal conductivity of the brass walls, no appreciable temperature variation was assumed to exist, radially, through the tube walls. The quantity of heat stored in the air column and in the Plexiglas plugs was considered negligible, compared to that in the brass walls, and was not included (37). A mean air temperature was computed as the arithmetic average of the measured wall temperatures.

Ten frequencies, spaced throughout the available range of the oscillator, were preselected for testing. During each test, a constant output from motor-generator to heater was maintained. Compensations for occasional slight variations in the output of the motor-generator were made by adjusting the rheostat. After the heater had been turned on, periodic readings were taken of the wall temperatures, as indicated by the tube-wall thermocouples, until the rate of increase of the temperature of both walls was observed to be approximately constant. Readings were then taken regularly, in most cases at fifteen minute intervals, of the four thermocouples in each wall and of the elapsed time between readings. The wall temperature, in this and all subsequent cases, was calculated as the equivalent in Fahrenheit degrees of the average millivolt readings of the four thermocouples in each wall. Several sets of readings were taken under these conditions, while a constant rate of increase of wall temperatures prevailed, thus providing the data for Case A.

With conditions otherwise unchanged, the oscillator was then turned on at a predetermined frequency. The signal was adjusted until an undistorted sine wave appeared on the oscilloscope screen, indicating that the signal was in resonance with the apparatus. At most frequencies, a large amplitude setting on the oscillator produced flat tops on the sine wave signal shown on the oscilloscope screen, which apparently indicated non-linearity of the current (38). In such cases, the amplitude was reduced until the distortion disappeared. The signal was then maintained at a condition of resonance, as closely as possible, for approximately one and a half hours at the same frequency, while tube wall temperatures were recorded approximately four times per hour. After sufficient points had been obtained to establish a constant rate of increase of the wall temperature of both walls, thus providing the data for Case C, the oscillator was turned off and several readings were recorded to insure that the rate of temperature increase tended to resume conditions comparable with those at the start of the test. A summary of the average temperature measurements and other data establishing the results for Case A and Case C are listed in Tables 1 through 10, Appendix C.

Computation of Driver Heat Input.--Five separate tests were conducted to determine the approximate amount of heat generated by the driver alone, at various frequencies in the range used during actual tests. The data obtained formed the basis for computations in Case B. Since the amount of energy generated by the driver depends upon the geometry and configuration of the system, the frequency transmitted, and the resonant response of the apparatus, no practical method could be devised by which to predetermine this quantity of heat by electrical measurements.

For these reasons, driver input calibration tests were conducted separately, duplicating the conditions of the actual runs except that there was no input to the heater. Each test was started after comparatively stable temperatures prevailed, initially, in the apparatus. With the heater off, several measurements were made of the wall temperatures, to establish the undisturbed rate of change of the wall temperatures. When this had been accomplished, the driver was turned on at the desired resonant frequency, and measurements were made every fifteen minutes for about one and a half hours, approximating the testing time for Case C data. The assumption was made that, as the driver began to introduce heat into the system, the heat was transmitted rapidly into the pipe walls, none being stored in the air annulus. Because of the thermal resistance created by the boundary conditions and the thick insulation at the outer surface of Tube B, and because of the relatively large conductivity of the brass walls, it was assumed further that, for a brief period of time, all of the heat generated by the driver was stored in the tube walls, causing a measurable rise in wall temperature, and that initially none flowed radially through the outer surface of Tube B.

Based on these assumptions, a value for the driver output during a specified interval of time was predicted. In all tests, except Test No. 2, the rate of change of the wall temperatures was negative prior to turning the driver on (heat stored in the test section was flowing into the insulation); in all cases the rate began to increase rapidly shortly after the driver was turned on, and reached a maximum after which it began to decrease (see Figure 7). Since there was no other input of heat during the test, the driver was obviously supplying energy to be stored in the tube walls at a rate sufficient to change the temperature of the walls from their initial temperatures to the maximum temperatures attained. This rate of heat storage was computed and transformed into a rate of heat generation of the driver, which can logically be assumed to prevail throughout any period of time that the driver remains in resonance at that frequency. Data from the driver input tests are summarized in Table 15, and the results are shown graphically in Figure 8.

It was predicted that the frequency response of the driver in decibels would fluctuate considerably over the range of frequencies tested, for reasons discussed in the next chapter. Although such fluctuations are suggested by the results, no definite trend was indicated in these tests relative to a relationship between the magnitude of the frequency and the magnitude of the maximum response. A constant frequency response appeared to be the best estimate of the conditions which occurred throughout the range of tests. Hence, an average of the maximum results from Tests D-2 through D-5 was obtained, as the most nearly representative figure to apply as a correction factor for the driver input, and this average value was assumed to be a constant value for the input of the driver at all frequencies in the test range. The omission of Test D-1 in establishing this average will be discussed in the next chapter.

The variation in the maximum value for S obtained from the various calibration tests may be attributed partially to several other variable factors, such as the difficulty experienced, especially at high frequencies, in maintaining resonance throughout the test; and temporary small variations in the output of the heater.

The decrease in the value of S after the maximum is attained, such as is reflected in the graphical summary of the tests (Figure 7), is to be expected, of course. The assumed conditions of zero heat flow, initially, through the outer wall actually could prevail only instantaneously, if at all, as the wall temperatures increase. The rate of flow through the outer surface of Tube B could be expected to increase continuously thereafter, thereby lowering the proportion of the generated heat which is reflected as heat stored in the walls.

This method of predicting the heat generation by the driver is certainly an approximation, but the estimate appears to be plausible. It is also important to consider that the estimate is modest in the amount predicted. Heat storage in the air column has been assumed negligible, although there would actually be an increasing amount of heat stored in the air column as the wall temperatures rise. Heat storage in and leakage through the end plugs, although doubtless considerably less than the heat stored in the brass walls, would consume additional heat which was provided by the driver but not measured. Perhaps of considerably larger magnitude is the effect already mentioned of assuming that no heat flows through the outer wall during the fifteen minutes to half an hour required to attain a peak in the rate of change of wall temperature; there is doubtless some transfer after the first increment of temperature rise in the walls, so that the "maximum" actually determined in each test may be considerably low. The effect of assuming that the magnitude of S is constant at 2.75 B/hr., for all frequencies, is discussed in more detail in the following chapter.

CHAPTER IV

DISCUSSION OF RESULTS

A summary of the data and results obtained in each of the ten test runs appears in Tables 1 through 10. The tests which were conducted with water flowing in the water jacket are summarized in Tables 11 through 13, for the purposes of comparison. As has been indicated, computations were carried out according to the formulas and procedures outlined in Figure 6, using the values for the physical characteristics as listed in Appendix E.

A value of the heat transfer coefficient at each wall was obtained for the conditions which existed during a period of relatively constant temperature change, under the conditions of both Case A and Case C. The two values were then averaged to obtain a single value for the coefficient before vibrations were introduced, and another value during the time in which vibrations were being induced. The per cent increase in the value of h_{av} after vibrations were introduced indicates the overall effect of the vibrations on the heat transfer coefficient in the air annulus.

The assumptions upon which the computations were based made necessary the employment of this average of the values of the coefficients at either wall of the air annulus. A mean value of the air temperature was assumed to exist in the air annulus, although the temperature of the air will obviously be higher next to the heated wall than it will be at the outer surface of the annulus. As can be seen from Figure 6, the values of h were obtained from the generalized formula

$$hA = \frac{q}{t_w - t_m},$$

where t_w is the temperature of the wall surface and t_m is the mean temperature of the air. Apparently, therefore, the quantity $(t_1 - t_m)$ in actuality will be smaller than that computed, since the air temperature is higher than the value of t_m which was used to approximate it; likewise, the value of $(t_m - t_3)$ will be slightly smaller than the values computed. An average of the two computed values of h, however, will largely compensate for the inaccuracies in the magnitudes of the individual coefficients which are introduced by the assumption of a mean air temperature.

The relationship between the per cent of increase in h_{av} and the frequency used during each test is indicated graphically in Figure 9. This figure indicates a maximum increase of the coefficient due to the introduction of vibrations into the system of 12 per cent, at a frequency of about 2,500 cycles per second (c.p.s.). There also appears to be a definite trend of the resultant effect of vibrations to increase from about five per cent at very low frequencies to the maximum mentioned, after which the effect becomes less as the frequency increases. This effect was predicted, since it was expected that the output of the oscillator would drop at the higher frequencies (see, for instance, 39, 40). An increase in the frequency of the oscillations, however, produces more nodes along the annulus and, consequently, contributes to a larger increase in h_{av}. Hence, from zero to about 2,500 c.p.s., the output of the oscillator and the increase in frequency both contribute to increasing h_{av}. At frequencies higher than 2,500 c.p.s., the effect of the higher frequencies in increasing h_{av} is offset by the decreasing intensity of the signal from the driver. The deviation of the results of Test No. 6 is discussed below.

As indicated in the discussion of the methods considered for the conduct of tests, the three tests made with water flowing in the water jacket are believed to be less dependable than those without water, due to fluctuations in the water temperature. Despite the limited range of frequencies attempted with water flowing in the jacket, the results are plotted in Figure 9, for the purposes of comparison. An examination of the results of Tests 11, 12, and 13 will indicate that the points also appear to bear out the trend toward an increase of the resultant effect on h_{av} from low frequencies to, in the case of these three tests, slightly over 300 c.p.s. The results of these three tests are, however, consistently lower than those obtained without water. No explanation is offered for this variation, except the general unreliability of the water-jacket tests, since fluctuations in the water temperature tend to mask the true rate of temperature change of Tube B.

The curve established by plotting the variations in h_{av} against the frequency is not as conclusive as was desired. There are several possible explanations for the fluctuations which occur in this curve.

The performance of the driver unit has a definite bearing upon the type of signal which was being transmitted into the air annulus.

Performance data and characteristics of the driver unit used were not available, but reference to any of several general discussions on the performance of commercial moving-coil speakers (for example, 40; 41; 42) indicates that the frequency response for such a device may vary widely over a comparatively small range of frequencies. Pender and McIlwain (43) present a curve for which the relative response in decibels of a "typical direct radiator speaker" is plotted against the frequency transmitted. From a maximum value of about 35 decibels, within a frequency range of 20,000 c.p.s., the curve fluctuates as much as 10 decibels in 500 c.p.s. The same discussion points out that such speakers are usually built to provide an optimum response in the middle frequency range (200 to 2,000 c.p.s.) for better reproduction of sounds usually transmitted over loud-speakers in practice. Manufacturer's data for new permanent-magnet driver units advertise accurate frequency reproduction in the general range 80 to 10,000 c.p.s. Amos and Kellaway (41) state that a response within plus or minus seven decibels over a range of 30 to 12,000 c.p.s. may be regarded as fairly good performance for a commercial loud-speaker. Loud-speaker performance, however, is measured in the open air. In this experiment, where the energy output of the driver was confined to a small, closed air column, there may be considerable variation from predicted loud-speaker performance.

The type of typical performance described, nevertheless, indicates that considerable variation should be expected in the results obtained in this experiment. The effect on h_{av} in Test No. 6 does not conform to the general trend of the performance curve of the other results, and it is suggested that this deviation may be caused by a low point in the output of the driver used. Attention is also invited to the performance obtained at very low frequencies. At 137 c.p.s., driver input calibration test D-1 reflected the lowest output of all frequencies tested. Likewise, tests No. 3 and 12, at the same frequency, indicate low magnitudes of the effect upon h_{av}, although the value of S which has been assumed as constant at all frequencies was used in calculating these values. With these facts in mind, and considering the typical ranges of response indicated by the references cited above, the conclusion is indicated that 137 c.p.s. is outside the range of accurate frequency response of the driver used, and the value obtained for S from test D-1 was not included in computing the average value for S.

Significant also is the fact that, for frequencies less than about 387 c.p.s., there would be no nodes established along the air annulus walls. For a node to be established, the length of the annulus must be greater than half a wave length. The velocity of sound is estimated to be approximately 1160 feet per second at 100°F. (44), in an air annulus about 18 in. long. The velocity is equal to the product of the frequency and the wave length; hence,

$$f = \frac{1160 \times 12}{18} = 775 \text{ c.p.s.}$$

The frequency for which the length of the chamber equals half a wave length, therefore, is 775/2 or about 390 c.p.s. This would be cause for predicting a low value for the per cent variation in h_{av} at frequencies below 390 c.p.s., which trend is borne out by the results.

Hence, with the exception of Test No. 6, the results indicate a characteristic which might be predicted. The output of the driver is probably low at low frequencies, rising to a peak somewhere near 2,500 c.p.s. At higher frequencies, the output of the speaker very likely decreases considerably, this effect being partially compensated for by the increased effect on the value of h_{av} as more nodes are created along the walls of the annulus at higher frequencies.

As has been indicated, the assumption that the output of the driver is a constant value, and the method used in establishing the value of that constant, admittedly may reduce the validity of the actual magnitudes established for h_{av} and h^*_{av}. However, any error introduced by this assumption must contribute toward decreasing the magnitude of the results obtained, rather than increasing them. The estimate was purposely established at the lowest value which seemed plausible, so that a faulty estimate could not indicate a rise in the value of h when one did not exist in fact. The estimated value of S might, conceivably, be as much as 100 per cent low, in which case the per cent increase for Test No. 5 would be about 18 per cent, instead of 12 per cent. Thus, although the actual increase in values of h_{av} may, conceivably, be higher than those indicated in the results, especially in the optimum frequency range of the driver, a consistent effort has been exerted throughout the assumptions needed for computations to assure that the results will not reflect values in excess of the actual case. It is therefore proposed that, while the assumptions may detract from the validity of the absolute values shown as results of the tests, they do not challenge the validity of the conclusions.

The results are rather sensitive to the values used for the physical properties involved in the transfer of heat during testing. Original computations involving areas, volumes, and heat capacities were refined to reflect as accurately as possible the true values involved, when this fact became apparent.

The dependence of the data upon a condition of resonance may explain some of the fluctuations in the results. Especially at high frequencies, repeated adjustment of the oscillator was necessary in order to maintain resonance. When the frequency varied from that which established a resonant condition, the effect of the vibrations would be lessened until the oscillator was readjusted.

Variation from the condition of resonance should have an additional influence on the validity of the value used as the output of the driver. That estimate, of course, is based upon the existence of the conditions of resonance. During the brief periods when the frequency was slightly out of resonance with the system, the input from the driver would be reduced, since the driver output will be a maximum when resonance prevails.

Similarly, but probably with less overall effect, there were temporary fluctuations in the input of the heater. When the motor-generator output varied slightly, compensation had to be applied by means of the rheostat, to reestablish the desired output to the heater.

The methods of measuring temperatures are believed to be within the desired accuracy of $0.10^{\circ}F$. The potentiometer was actually read to an equivalent accuracy of about $0.01^{\circ}F$.

CHAPTER V

CONCLUSIONS

As a result of the experiments which have been described, the following conclusions are drawn:

1. Using typical laboratory equipment, this experiment has indicated that an increase occurs in the heat transfer coefficient when a standing acoustic wave is introduced into a closed air column.

2. The maximum increase of the coefficient was slightly larger than ten per cent. The results obtained represent a probable lower limit to the actual effect obtained.

3. The effect of vibration on the heat transfer coefficient, using the equipment described, increases from the lowest frequency investigated to a frequency of about 2500 c.p.s., after which it decreases with increasing frequency.

4. More accurate measurement of the quantity of heat transmitted to the air column by the driver, and of the general performance and frequency response of the driver, is essential to obtaining more conclusive results.

CHAPTER VI

RECOMMENDATIONS

It is recommended:

1. That further tests be conducted to verify the value of the magnitude of the effect of vibrations on the heat transfer coefficient. Although many types of apparatus are possible for this purpose, this type of apparatus appears to be worthy of refinement, since it possesses the advantage of confining the sonic energy of the driver to the system, and of isolating the system from the effects of convective currents, unpredictable variations in the medium, and other external influences.

2. That the amplitude and frequency response characteristics of any speaker used to transmit vibrations into the column be thoroughly investigated to determine the consistency of its output and the optimum frequency range for its operation. Tests should then be restricted to frequencies at which the speaker provides accurate and consistent frequency response. In addition, some means should be devised to accurately measure the speaker output while the test is in progress. It is suggested that a small microphone could be introduced through the side of the lower Plexiglas plug to make possible the measurement of the speaker output in decibels.

3. That any future experiments of this type be conducted in a room which can be maintained at constant temperature, if possible. In this case, the insulation around the apparatus should be reduced

considerably, so that steady state conditions would prevail quickly after a change of temperature in the test section. An alternate method of accomplishing this end would be the provision of a fluid for the water jacket which was not subject to changes and fluctuations in temperature.

4. That a source of variable, direct-current power, which can be maintained without adjustment for long periods of time with a minimum of fluctuation be obtained, if possible. This source should be capable of producing power in excess of that needed during the actual tests, in order to decrease the warm-up time of the apparatus. Each series of tests should be conducted with the same heat input from the heater, to eliminate one more variable factor during comparison of the results obtained.

5. That a method be devised to reduce fluctuations of the frequency output of the oscillator during the testing period. A convenient method of checking the frequency might be the use of an electronic switch in conjunction with the oscilloscope, impressing a standard frequency wave on the screen to be used as a check on the test signal. It may be desirable to investigate the possibility of using a piezoelectric crystal or magnetostriction oscillator.

6. That measurements of the temperature be made at some point close to the outer wall of the test section, but protected from fluctuations in environmental temperature, if the water jacket is not used. This would permit accumulation of data which could serve as a check on the flow of heat through the air column by providing data for an additional heat balance.

7. That any future experiments include a comparison of the effects produced with and without resonance, at approximately the same frequency and at the same amplitude of the signal. It appeared possible during the period in which tests were conducted, although no formal tests were carried out to verify the possibility, that the rate of change of the wall temperatures increased disproportionately when the frequency was varied from one point of resonance to another. It would be interesting to explore the possibility that continuous variation of the frequency through points of resonance within the optimum frequency range of the equipment might produce a larger effect on h_{av} than tests conducted at a single resonant frequency. The only explanation which suggests itself to predict such a result is the possibility that this method of conducting the test might contribute to additional turbulence in the boundary layer by, in effect, continuously moving the nodes along the surface of the annular walls.

APPENDIX A

LIST OF SYMBOLS

LIST OF SYMBOLS

Symbol	Meaning
amp.	ampere
B	British Thermal Unit
A	area
c_p	specific heat, B/lb_m-^oF
c.p.s.	cycles per second
^oC.	degrees Centigrade
db.	decibels
D	diameter
e.m.f.	electromotive force
f	frequency
ft.	feet
^oF.	degrees Fahrenheit
g	acceleration due to gravity
g.p.m.	gallons per minute
Gr	Grashof number
h	heat transfer coefficient
h*	heat transfer coefficient with vibrational influence
hr.	hours
H	height
in.	inches
ID	inner diameter
k	thermal conductivity, B/hr-ft-^oF.
k_c	equivalent thermal conductivity, including effects of convection
lb_m	pounds mass
ln	natural logarithm
L	length
L & N	Leeds and Northrup Co.
mv	millivolts
No.	number
OD	outer diameter
pp.	pages
q	heat flow, B/hr.
^oR	degrees Rankine
S	heat input of driver, B/hr.
t	temperature
T	absolute temperature
v	volts
V	volume, ft^3
w	watts
x	distance, ft.
Tube A	inner brass pipe
Tube B	outer brass pipe

Greek Letters

β	coefficient of expansion
θ	time, hr.
π	3.1416
ρ	density, lb_m/ft^3
μv	microvolts
ν	kinematic viscosity
Δ	change in __ __

Subscripts

1	OD, Tube A
2	annulus
3	ID, Tube B
a	Tube A
av	average of values at D_1 and D_3
b	Tube B
d	driver chamber
e	heater end space
h	heater
m	mean
p	at constant pressure
pl	hemispherical plug
t	test section annulus
w	wall
x	heater air annulus

APPENDIX B

FIGURES

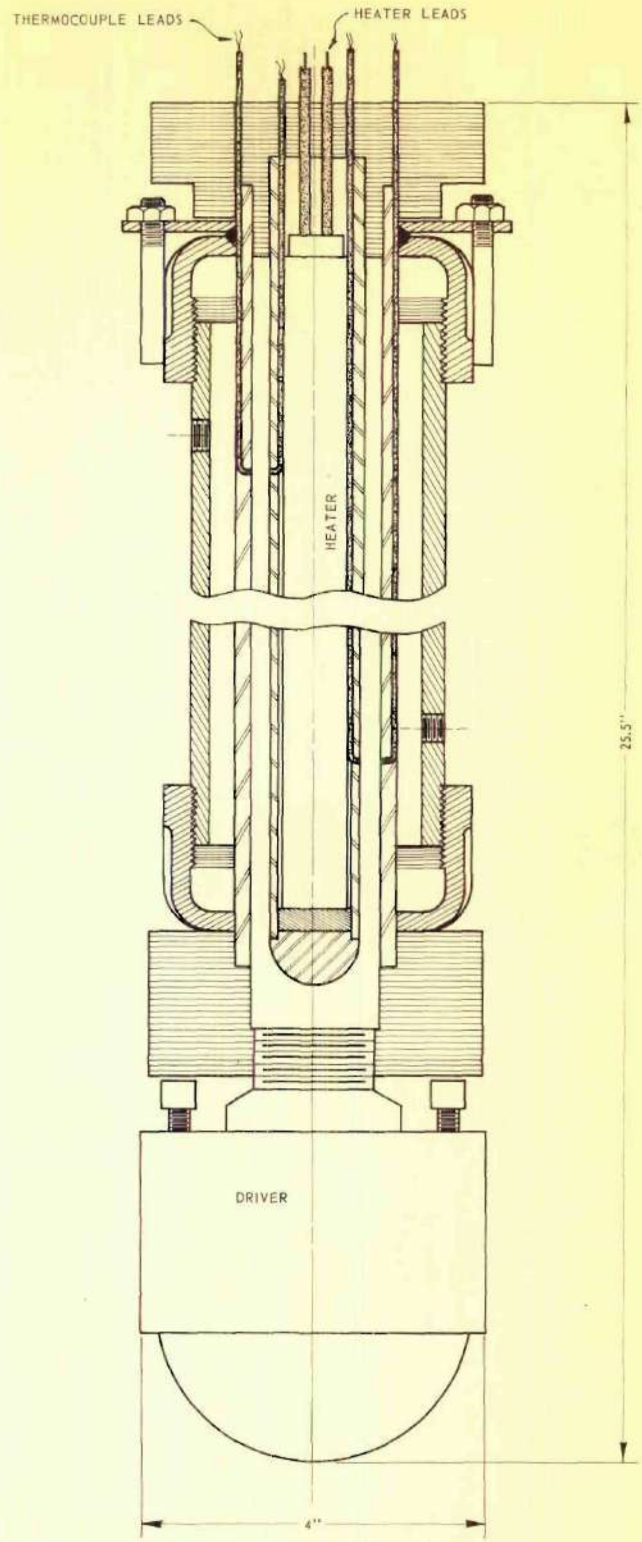


FIGURE 1. ASSEMBLY DRAWING OF APPARATUS.

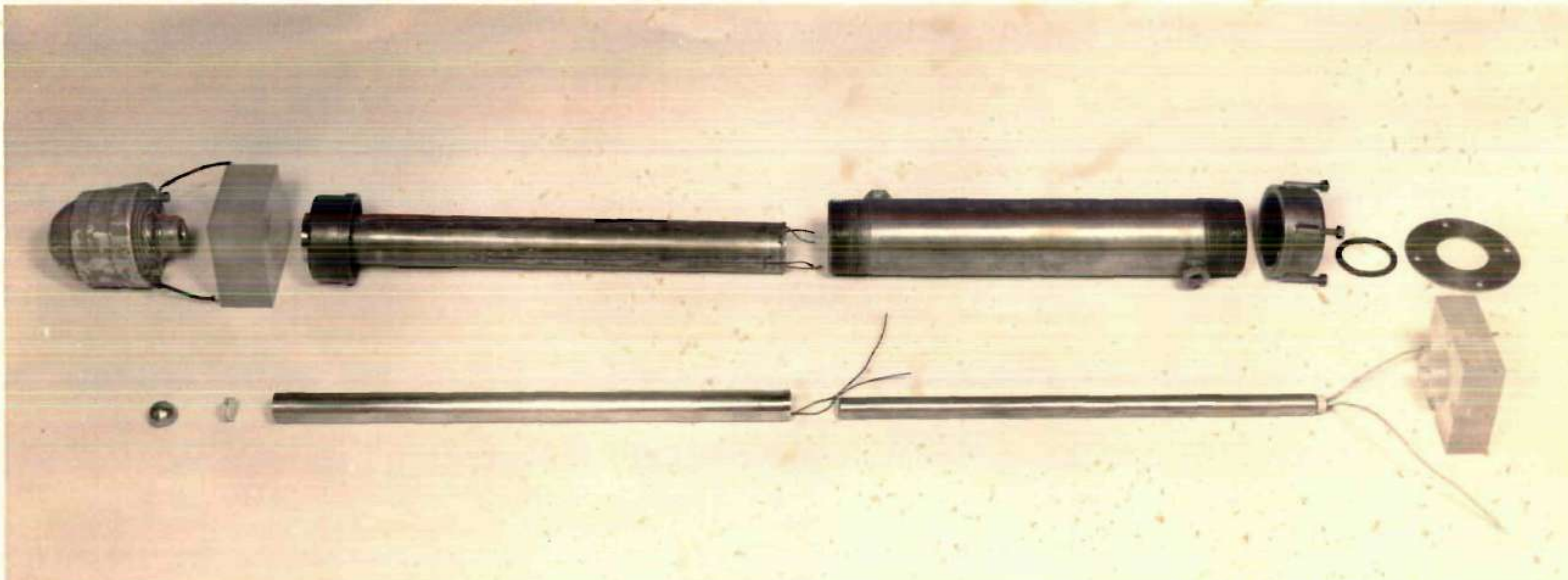


FIGURE 2. PHOTOGRAPH OF APPARATUS, DISSASSEMBLED AFTER COMPLETION OF TESTING.

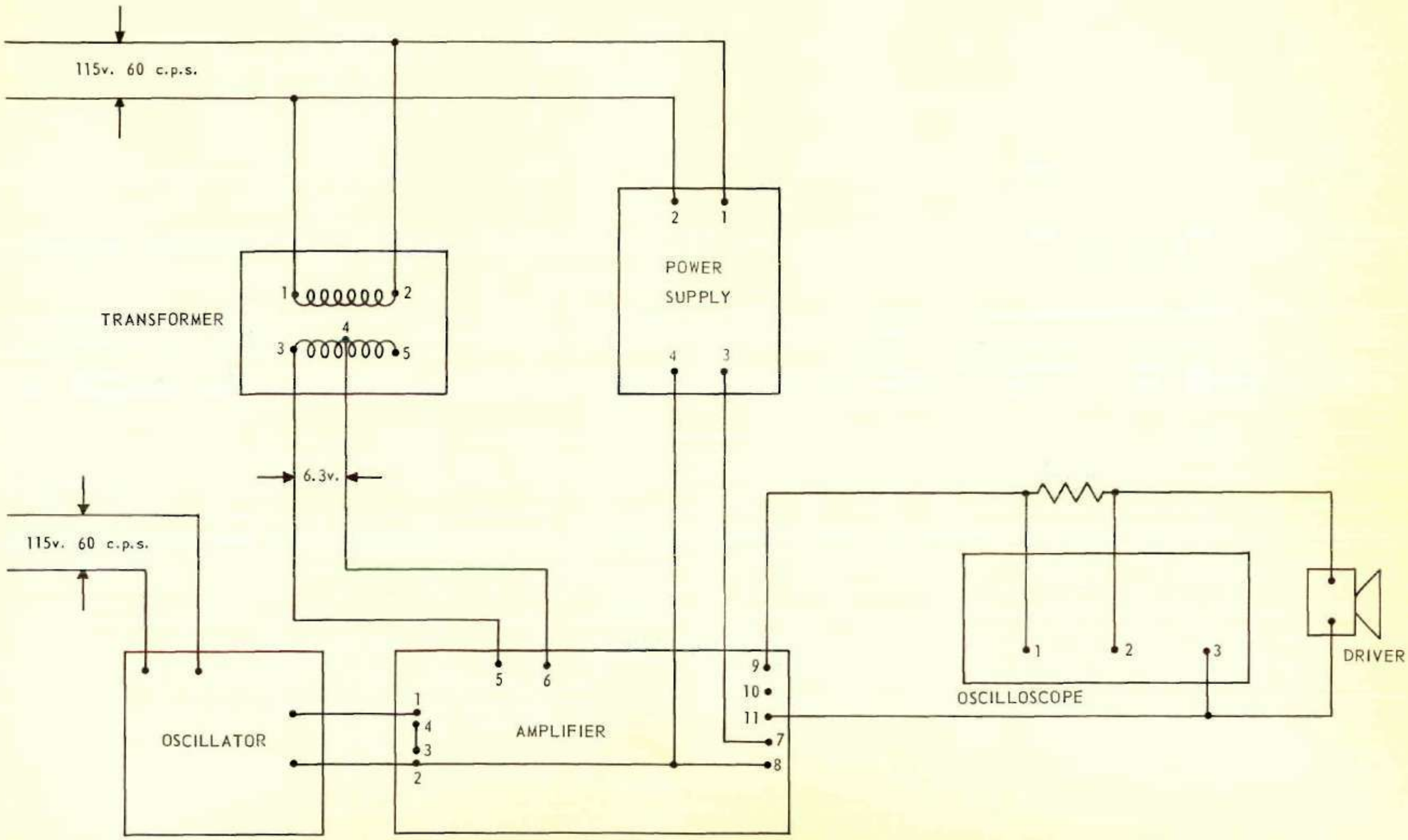


FIGURE 3. WIRING DIAGRAM FOR OSCILLATOR CIRCUIT.

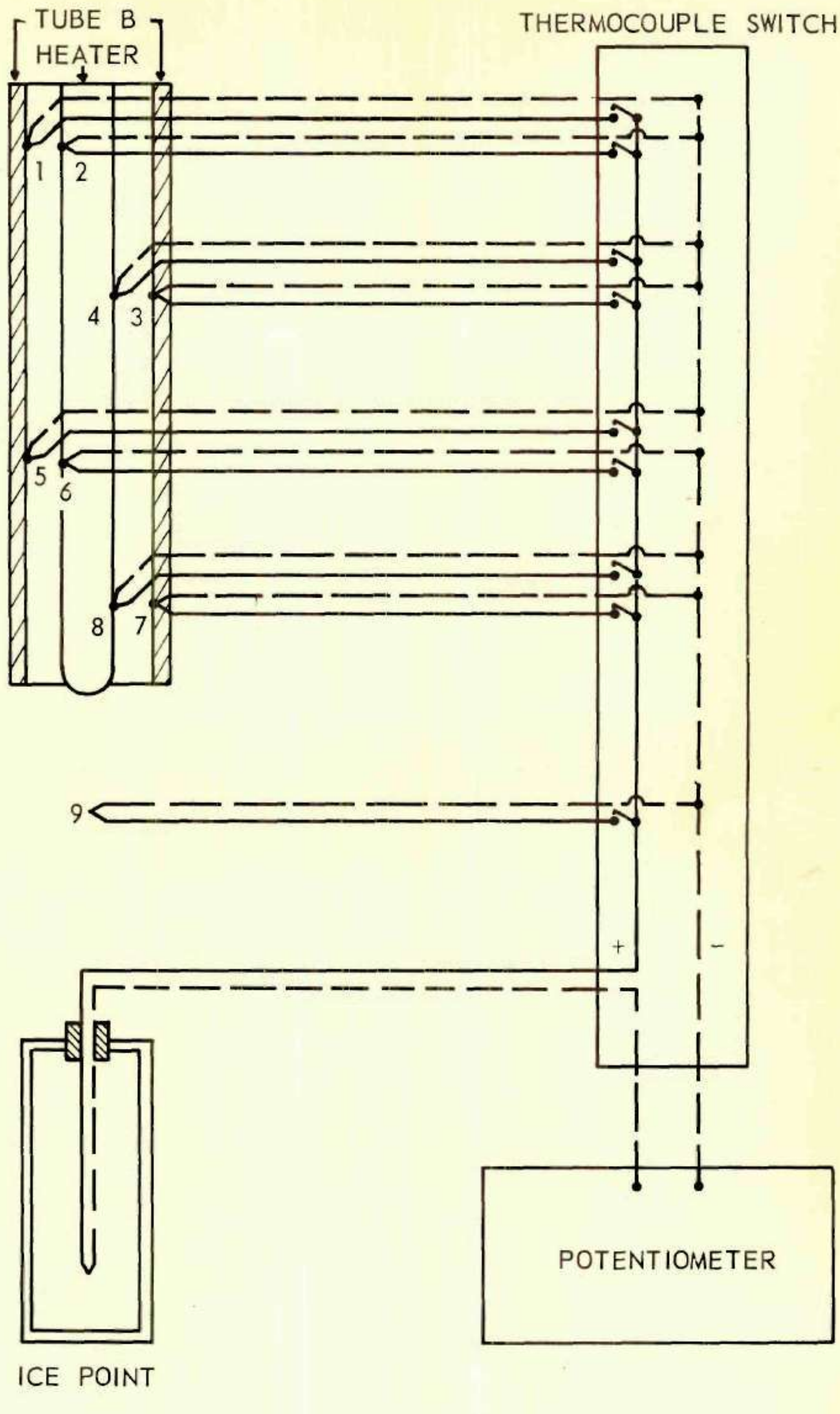
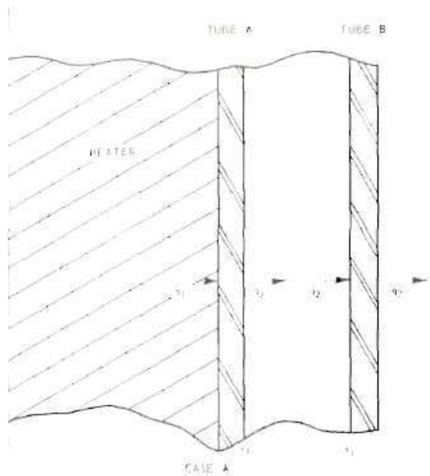


FIGURE 4. THERMOCOUPLE WIRING DIAGRAM.

FIGURE 5. PHOTOGRAPH OF INSTRUMENTS AND EQUIPMENT USED IN TESTING.



CASE A

NEGLECT HEAT STORAGE IN THE AIR COLUMN.

q_1 = heater output, B/hr

$$q_1 - q_2 = (c\rho V)_1 \frac{\Delta t_1}{\Delta\theta} = 0.327 \frac{\Delta t_1}{\Delta\theta}; \quad q_2 = q_1 - 0.327 \frac{\Delta t_1}{\Delta\theta}, \ \frac{B}{hr}$$

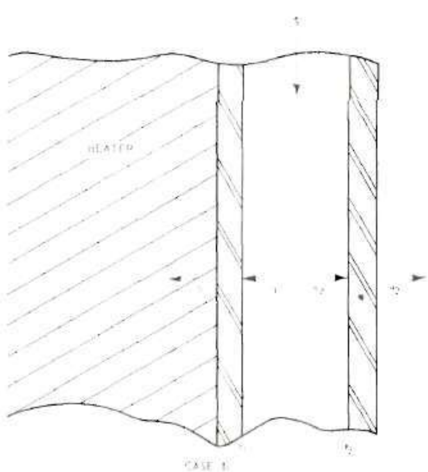
$$t_m = \frac{t_1 + t_3}{2}, \ {}^{\circ}F$$

$$h_1 = \frac{q_2}{A_1(t_1 - t_m)} = \frac{2q_2}{0.407(t_1 - t_3)} = \frac{q_2}{0.204(t_1 - t_3)}, \ \frac{B}{hr\text{-}ft^2\text{-}{}^{\circ}F}$$

$$h_3 = \frac{q_2}{A_3(t_m - t_3)} = \frac{2q_2}{0.577(t_1 - t_3)} = \frac{q_2}{0.289(t_1 - t_3)}, \ \frac{B}{hr\text{-}ft^2\text{-}{}^{\circ}F}$$

$$q_2 - q_3 = (c\rho V)_3 \frac{\Delta t_3}{\Delta\theta}, \ \frac{B}{hr}$$

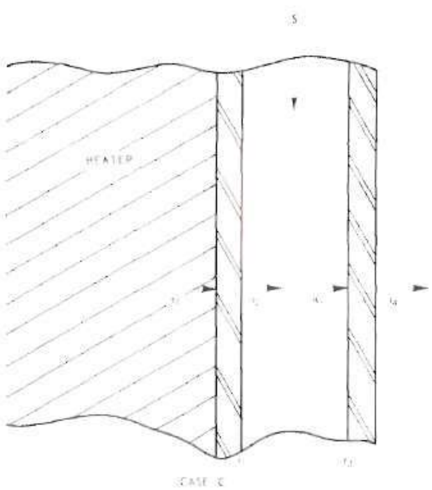
$$q_3 = q_2 - 0.557 \frac{\Delta t_3}{\Delta\theta}, \ \frac{B}{hr}$$



CASE B

$S = q_1 + q_2$, B/hr, initially

$$= 0.327 \frac{\Delta t_1}{\Delta\theta} + 0.557 \frac{\Delta t_3}{\Delta\theta} = 2.75 \ \frac{B}{hr}$$



CASE C

$$q_1 - q_2 = (c\rho V)_1 \frac{\Delta t_1}{\Delta\theta} = 0.327 \frac{\Delta t_1}{\Delta\theta} \ \frac{B}{hr}$$

$$q_2 = q_1 - 0.327 \frac{\Delta t_1}{\Delta\theta} \ B/hr$$

$$q_3 = q_2 + S \ B/hr$$

$$q_3 - q_4 = (c\rho V)_3 \frac{\Delta t_3}{\Delta\theta} = 0.557 \frac{\Delta t_3}{\Delta\theta} \ \frac{B}{hr}$$

$$q_4 = q_3 - 0.557 \frac{\Delta t_3}{\Delta\theta}$$

$$h_1^* = \frac{q_2}{A_1(t_1 - t_m)} = \frac{q_2}{0.204(t_1 - t_3)} \ \frac{B}{hr\text{-}ft^2\text{-}F}$$

$$h_3^* = \frac{q_3}{A_3(t_m - t_3)} = \frac{q_3}{0.289(t_1 - t_3)} \ \frac{B}{hr\text{-}ft^2\text{-}F}$$

FIGURE 6. FORMULAS FOR HEAT FLOW THROUGH TEST SECTION WALLS UNDER THE CONDITIONS OF CASE A, CASE B, AND CASE C.

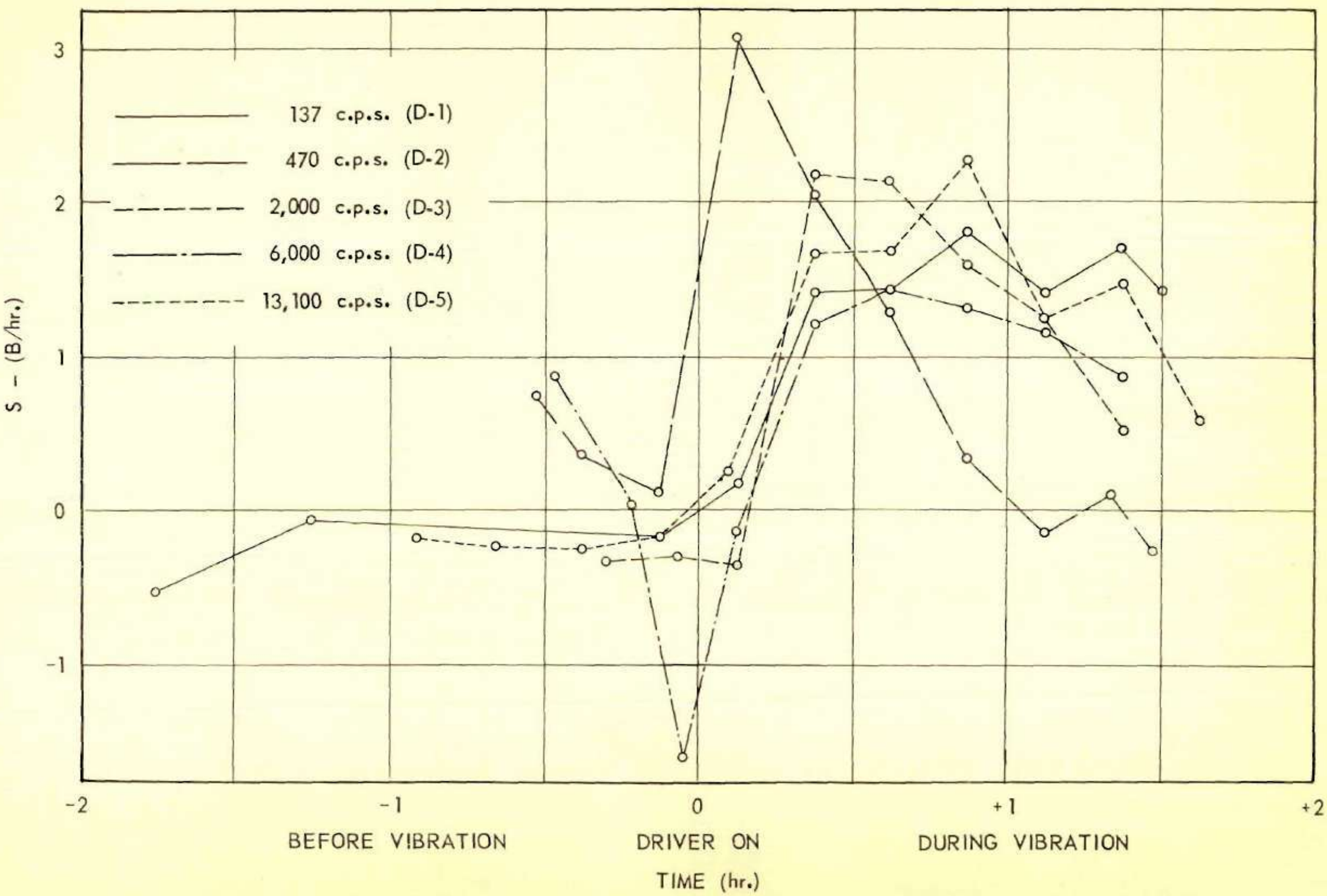


FIGURE 7. DRIVER INPUT CALIBRATION TESTS D-1 THROUGH D-5. HEAT INPUT OF DRIVER DURING TEST PERIOD.

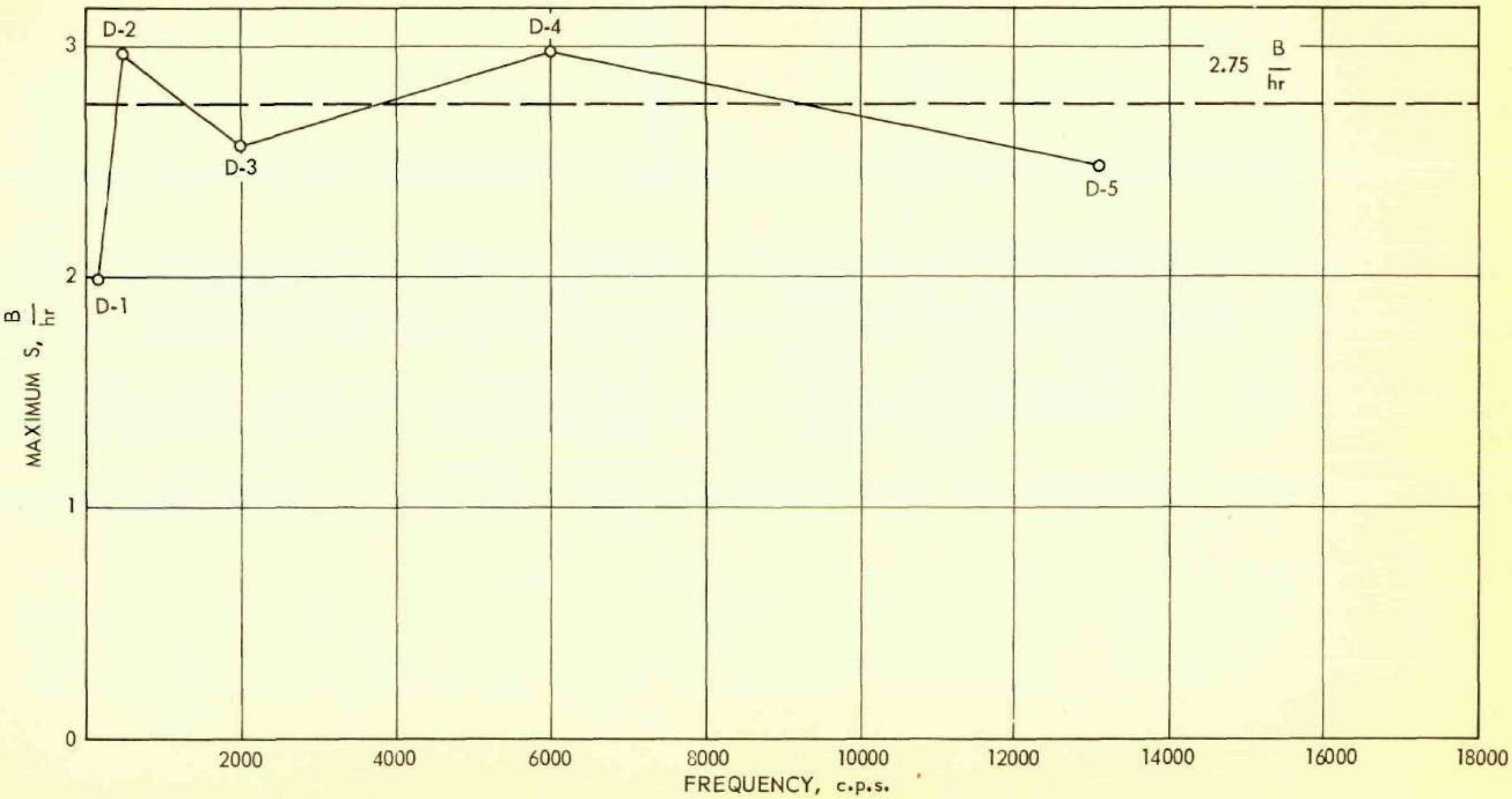


FIGURE 8. DRIVER INPUT CALIBRATION TESTS D-1 THROUGH D-5. MAXIMUM HEAT INPUT OF DRIVER AT THE FREQUENCIES TESTED.

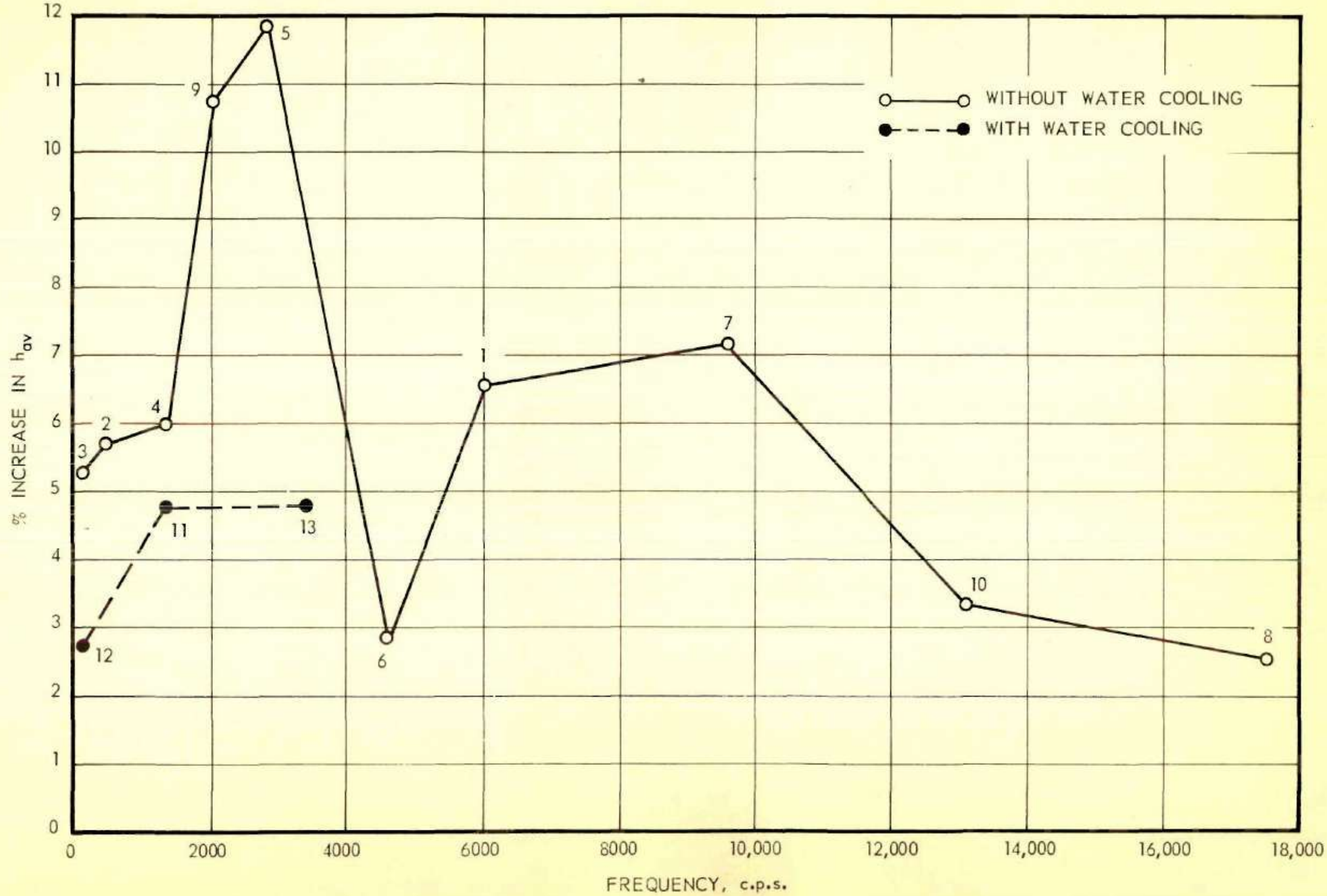


FIGURE 9. PER CENT INCREASE IN H_{av} DUE TO ACOUSTIC VIBRATION AT THE FREQUENCIES TESTED (TESTS NO. 1 THROUGH 13.)

APPENDIX C

TABLES

Table 1. Heat Transfer Data

q_1: 12.16 B/hr. f: 6000 c.p.s.

S: 2.75 B/hr. Run No. 1

		Before Vibration, Case A		During Vibration, Case C
$\Delta\Theta$	hr.	.5		.916
Δt_1	^{o}F.	.6		2.6
$\Delta t_1/\Delta\Theta$	^{o}F/hr.	1.2		2.84
$.327\ \Delta t_1/\Delta\Theta$	^{o}F/hr.	.393		.928
q_2	B/hr.	11.767		11.23
Δt_3	^{o}F.	.6		2.6
$\Delta t_3/\Delta\Theta$	^{o}F/hr.	1.2		2.84
$.557\ \Delta t_3/\Delta\Theta$	^{o}F/hr.	.669		1.582
q_3	B/hr.	11.098		13.98
$(t_1)av$	^{o}F	113.10		115.93
$(t_3)av$	^{o}F	92.87		95.93
$(t_1)av-(t_3)av$	^{o}F	20.23		20.00
h_1	B/hr-ft^2-^{o}F	2.85	h_1*	2.75
h_3	B/hr-ft^2-^{o}F	2.01	h_3*	2.42
h_{av}	B/hr-ft^2-^{o}F	2.43	h_{av}*	2.59

% increase in h_{av}: 6.58

Table 2. Heat Transfer Data

q_1: 12.63 B/hr. f: 470 c.p.s.

S: 2.75 B/hr. Run No. 2

		Before Vibration, Case A		During Vibration, Case C
$\Delta\Theta$	hr.	.5		1.333
Δt_1	^oF	-.05		3.1
$\Delta t_1/\Delta\Theta$	$^oF/hr$	-.10		2.325
$.327\ \Delta t_1/\Delta\Theta$	$^oF/hr$	-.033		.76
q_2	B/hr	12.663		11.87
Δt_3	^oF	-.1		2.15
$\Delta t_3/\Delta\Theta$	$^oF/hr$	-.2		1.612
$.557\ \Delta t_3/\Delta\Theta$	$^oF/hr$	-.111		.897
q_3	B/hr	12.774		14.62
$(t_1)av$	^oF	118.18		120.63
$(t_3)av$	^oF	98.15		101.14
$(t_1)av-(t_3)av$	^oF	20.03		19.49
h_1	$B/hr\text{-}ft^2\text{-}^oF$	3.09	h_1*	2.98
h_3	$B/hr\text{-}ft^2\text{-}^oF$	2.18	h_3*	2.60
h_{av}	$B/hr\text{-}ft^2\text{-}^oF$	2.64	h_{av}*	2.79

% increase in h_{av}: 5.68

Table 3. Heat Transfer Data

q_1: 16.66 B/hr. f: 137 c.p.s.

S: 2.75 B/hr. Run No. 3

		Before Vibration, Case A		During Vibration, Case C
$\Delta\Theta$	hr	1		1
Δt_1	^{o}F	1.8		2.7
$\Delta t_1/\Delta\Theta$	$^{o}F/hr$	1.8		2.7
$.327\ \Delta t_1/\Delta\Theta$	$^{o}F/hr$	.588		.883
q_2	B/hr	16.072		15.777
Δt_3	^{o}F	1.6		1.95
$\Delta t_3/\Delta\Theta$	$^{o}F/hr$	1.6		1.95
$.557\ \Delta t_3/\Delta\Theta$	$^{o}F/hr$	.892		1.087
q_3	B/hr	15.180		18.527
$(t_1)av$	^{o}F	124.74		128.5
$(t_3)av$	^{o}F	99.36		103.19
$(t_1)av-(t_3)av$	^{o}F	25.38		25.31
h_1	$B/hr\text{-}ft^2\text{-}^{o}F$	3.10	h_1*	3.05
h_3	$B/hr\text{-}ft^2\text{-}^{o}F$	2.19	h_3*	2.53
h_{av}	$B/hr\text{-}ft^2\text{-}^{o}F$	2.65	h_{av}*	2.79

% increase in h_{av}: 5.28

Table 4. Heat Transfer Data

q_1: 16:66 B/hr. f: 1320 c.p.s.

S: 2:75 B/hr. Run No. 4

		Before Vibration, Case A		During Vibration, Case C
$\Delta\Theta$	hr	.25		.5
Δt_1	^{o}F	.3		.8
$\Delta t_1/\Delta\Theta$	^{o}F/hr	1.20		1.60
.327 $\Delta t_1/\Delta\Theta$	^{o}F/hr	.393		.523
q_2	B/hr	16.267		16.137
Δt_3	^{o}F	.1		.85
$\Delta t_3/\Delta\Theta$	^{o}F/hr	.4		1.70
.557 $\Delta t_3/\Delta\Theta$	^{o}F/hr	.223		.947
q_3	B/hr	16.044		18.887
$(t_1)av$	^{o}F	130.25		130.73
$(t_3)av$	^{o}F	104.85		105.28
$(t_1)av-(t_3)av$	^{o}F	25.40		25.45
h_1	B/hr-ft^2-^{o}F	3.14	h_1*	3.11
h_3	B/hr-ft^2-^{o}F	2.21	h_3*	2.57
h_{av}	B/hr-ft^2-^{o}F	2.68	h_{av}*	2.84

% increase in h_{av}: 5.97

Table 5. Heat Transfer Data

q_1: 20.29 B/hr. f: 2480 c.p.s.

S: 2.75 B/hr. Run No. 5

		Before Vibration, Case A		During Vibration, Case C
$\Delta\Theta$	hr	.25		.5
Δt_1	^oF	.65		1.25
$\Delta t_1/\Delta\Theta$	$^oF/hr$	2.60		2.50
$.327\ \Delta t_1/\Delta\Theta$	$^oF/hr$	.850		.818
q_2	B/hr	19.440		19.472
Δt_3	^oF	.85		1.35
$\Delta t_3/\Delta\Theta$	$^oF/hr$	3.40		2.70
$.557\ \Delta t_3/\Delta\Theta$	$^oF/hr$	1.892		1.503
q_3	B/hr	17.548		22.222
$(t_1)av$	^oF	136.18		144.12
$(t_3)av$	^oF	101.78		111.45
$(t_1)av-(t_3)av$	^oF	34.40		32.67
h_1	$B/hr\text{-}ft^2\text{-}^oF$	2.77	h_1*	2.92
h_3	$B/hr\text{-}ft^2\text{-}^oF$	1.95	h_3*	2.35
h_{av}	$B/hr\text{-}ft^2\text{-}^oF$	2.36	h_{av}*	2.64

% increase in h_{av}: 11.87

Table 6. Heat Transfer Data

q_1: 20.29 B/hr. f: 4600 c.p.s.

S: 2.75 B/hr. Run No. 6

		Before Vibration, Case A		During Vibration, Case C
$\Delta\Theta$	hr	.583		1.33
Δt_1	$^{\circ}F$	.80		3.55
$\Delta t_1/\Delta\Theta$	$^{\circ}F/hr$	1.37		2.67
$.327\ \Delta t_1/\Delta\Theta$	$^{\circ}F/hr$	.448		.873
q_2	B/hr	19.842		19.417
Δt_3	$^{\circ}F$	.65		3.40
$\Delta t_3/\Delta\Theta$	$^{\circ}F/hr$	1.114		2.55
$.557\ \Delta t_3/\Delta\Theta$	$^{\circ}F/hr$	.621		1.424
q_3	B/hr	19.221		22.167
$(t_1)av$	$^{\circ}F$	147.52		150.76
$(t_3)av$	$^{\circ}F$	113.43		116.37
$(t_1)av-(t_3)av$	$^{\circ}F$	34.09		34.39
h_1	$B/hr\text{-}ft^2\text{-}^{\circ}F$	2.85	h_1*	2.77
h_3	$B/hr\text{-}ft^2\text{-}^{\circ}F$	2.01	h_3*	2.23
h_{av}	$B/hr\text{-}ft^2\text{-}^{\circ}F$	2.43	h_{av}*	2.50

% increase in h_{av}: 2.88

Table 7. Heat Transfer Data

q_1: 18.43 B/hr. f: 9600 c.p.s.

S: 2.75 B/hr. Run No. 7

		Before Vibration, Case A		During Vibration Case C
$\Delta\Theta$	hr	.25		1.167
Δt_1	$^\circ F$	.70		3.70
$\Delta t_1/\Delta\Theta$	$^\circ F/hr$	2.80		3.17
$.327\ \Delta t_1/\Delta\Theta$	$^\circ F/hr$	.916		1.038
q_2	B/hr	17.514		17.392
Δt_3	$^\circ F$	.85		3.30
$\Delta t_3/\Delta\Theta$	$^\circ F/hr$	3.40		2.83
$.557\ \Delta t_3/\Delta\Theta$	$^\circ F/hr$	1.892		1.577
q_3	B/hr	15.622		20.142
$(t_1)av$	$^\circ F$	126.80		133.02
$(t_3)av$	$^\circ F$	99.15		105.67
$(t_1)av-(t_3)av$	$^\circ F$	27.65		27.35
h_1	$B/hr\text{-}ft^2\text{-}^\circ F$	3.10	$h_1{}^*$	3.12
h_3	$B/hr\text{-}ft^2\text{-}^\circ F$	2.19	$h_3{}^*$	2.55
h_{av}	$B/hr\text{-}ft^2\text{-}^\circ F$	2.65	$h_{av}{}^*$	2.84

% increase in h_{av}: 7.18

Table 8. Heat Transfer Data

q_1: 18.43 B/hr. f: 17,500 c.p.s.

S: 2.75 B/hr. Run No. 8

		Before Vibration, Case A		During Vibration, Case C
$\Delta\Theta$	hr	.833		.5
Δt_1	$^{\circ}F$	1.15		1.30
$\Delta t_1/\Delta\Theta$	$^{\circ}F/hr$	1.38		2.60
$.327\ \Delta t_1/\Delta\Theta$	$^{\circ}F/hr$	.452		.851
q_2	B/hr	17.978		17.589
Δt_3	$^{\circ}F$	1.05		1.30
$\Delta t_3/\Delta\Theta$	$^{\circ}F/hr$	1.26		2.60
$.557\ \Delta t_3/\Delta\Theta$	$^{\circ}F/hr$	.702		1.448
q_3	B/hr	17.276		20.339
$(t_1)_{av}$	$^{\circ}F$	136.37		138.12
$(t_3)_{av}$	$^{\circ}F$	108.65		110.53
$(t_1)_{av}-(t_3)_{av}$	$^{\circ}F$	27.72		27.59
h_1	$B/hr\text{-}ft^2\text{-}^{\circ}F$	3.18	h_1*	3.13
h_3	$B/hr\text{-}ft^2\text{-}^{\circ}F$	2.25	h_3*	2.55
h_{av}	$B/hr\text{-}ft^2\text{-}^{\circ}F$	2.77	$h_{av}*$	2.84

% increase in h_{av}: 2.53

Table 9. Heat Transfer Data

q_1: 18.43 B/hr. f: 2000 c.p.s.

S: 2.75 B/hr. Run No. 9

		Before Vibration, Case A		During Vibration, Case C
$\Delta\Theta$	hr	.25		.75
Δt_1	^{o}F	.7		1.8
$\Delta t_1/\Delta\Theta$	$^{o}F/hr$	2.8		2.40
$.327\ \Delta t_1/\Delta\Theta$	$^{o}F/hr$	.916		.785
q_2	B/hr	17.514		17.645
Δt_3	^{o}F	.85		2.0
$\Delta t_3/\Delta\Theta$	$^{o}F/hr$	3.40		2.67
$.557\ \Delta t_3/\Delta\Theta$	$^{o}F/hr$	1.892		1.488
q_3	B/hr	15.622		20.395
$(t_1)_{av}$	^{o}F	127.50		135.88
$(t_3)_{av}$	^{o}F	99.38		108.54
$(t_1)_{av}-(t_3)_{av}$	^{o}F	28.12		27.34
h_1	$B/hr\text{-}ft^2\text{-}^{o}F$	3.05	h_1^*	3.17
h_3	$B/hr\text{-}ft^2\text{-}^{o}F$	2.15	h_3^*	2.58
h_{av}	$B/hr\text{-}ft^2\text{-}^{o}F$	2.60	h_{av}^*	2.88

% increase in h_{av}: 10.78

Table 10. Heat Transfer Data

q_1: 18.43 B/hr. f: 13,100 c.p.s.

S: 2.75 B/hr. Run No. 10

		Before Vibration, Case A		During Vibration, Case C
$\Delta\Theta$	hr	.67		.833
Δt_1	^{o}F	.8		2.4
$\Delta t_1/\Delta\Theta$	$^{o}F/hr$	1.194		2.882
$.327\ \Delta t_1/\Delta\Theta$	$^{o}F/hr$	.390		.942
q_2	B/hr	18.040		17.488
Δt_3	^{o}F	.65		2.15
$\Delta t_3/\Delta\Theta$	$^{o}F/hr$	.970		2.585
$.557\ \Delta t_3/\Delta\Theta$	$^{o}F/hr$	.541		1.439
q_3	B/hr	17.499		20.238
$(t_1)av$	^{o}F	138.37		140.80
$(t_3)av$	^{o}F	110.68		113.02
$(t_1)av-(t_3)av$	^{o}F	27.69		27.78
h_1	$B/hr\text{-}ft^2\text{-}{}^{o}F$	3.19	h_1*	3.09
h_3	$B/hr\text{-}ft^2\text{-}{}^{o}F$	2.25	h_3*	2.53
h_{av}	$B/hr\text{-}ft^2\text{-}{}^{o}F$	2.72	h_{av}*	2.81

% increase in h_{av}: 3.31

Table 11. Heat Transfer Data

q_1: 18.43 B/hr. f: 1320 c.p.s.

S: 2.75 B/hr. Run No. 11

		Before Vibration, Case A		During Vibration, Case C
$\Delta\Theta$	hr	.5		1.75
Δt_1	^oF	.5		4.15
$\Delta t_1/\Delta\Theta$	^oF/hr	1.00		2.37
$.327\ \Delta t_1/\Delta\Theta$	^oF/hr	.327		.775
q_2	B/hr	18.103		17.655
Δt_3	^oF	-.4		4.05
$\Delta t_3/\Delta\Theta$	^oF/hr	-.8		2.31
$.557\ \Delta t_3/\Delta\Theta$	^oF/hr	-.456		1.290
q_3	B/hr	18.559		20.405
$(t_1)av$	^oF	110.73		113.85
$(t_3)av$	^oF	80.92		83.99
$(t_1)av-(t_3)av$	^oF	29.81		29.86
h_1	B/hr-ft^2-^oF	2.98	h_1*	2.95
h_3	B/hr-ft^2-^oF	2.10	h_3*	2.37
h_{av}	B/hr-ft^2-^oF	2.54	h_{av}*	2.32

% increase in h_{av}: 4.72

Table 12. Heat Transfer Data

q_1: 18.43 B/hr. f: 137 c.p.s.

S: 2.75 B/hr. Run No. 12

		Before Vibration, Case A		During Vibration, Case C
$\Delta\Theta$	hr	.167		.25
Δt_1	^{o}F	.05		.15
$\Delta t_1/\Delta\Theta$	$^{o}F/hr$	.30		.60
$.327\ \Delta t_1/\Delta\Theta$	$^{o}F/hr$	.098		.196
q_2	B/hr	18.332		18.234
Δt_3	^{o}F	.15		.5
$\Delta t_3/\Delta\Theta$	$^{o}F/hr$	.90		2.00
$.557\ \Delta t_3/\Delta\Theta$	$^{o}F/hr$	.502		1.054
q_3	B/hr	17.830		21.484
$(t_1)av$	^{o}F	115.98		117.73
$(t_3)av$	^{o}F	86.23		86.75
$(t_1)av-(t_3)av$	^{o}F	29.75		30.98
h_1	$B/hr\text{-}ft^2\text{-}^{o}F$	3.02	h_1*	2.89
h_3	$B/hr\text{-}ft^2\text{-}^{o}F$	2.13	h_3*	2.40
h_{av}	$B/hr\text{-}ft^2\text{-}^{o}F$	2.58	h_{av}*	2.65

% increase in h_{av}: 2.71

Table 13. Heat Transfer Data

q_1: 18.43 B/hr. f: 3420 c.p.s.

S: 2.75 B/hr. Run No. 13

		Before Vibration, Case A		During Vibration, Case C
$\Delta\Theta$	hr	.583		1
Δt_1	^{o}F	.2		2.05
$\Delta t_1/\Delta\Theta$	$^{o}F/hr$	.343		2.05
$.327\ \Delta t_1/\Delta\Theta$	$^{o}F/hr$	.112		.671
q_2	B/hr	18.318		17.749
Δt_3	^{o}F	-.6		2.10
$\Delta t_3/\Delta\Theta$	$^{o}F/hr$	-1.03		2.10
$.557\ \Delta t_3/\Delta\Theta$	$^{o}F/hr$	-.573		1.17
q_3	B/hr	18.891		20.499
$(t_1)av$	^{o}F	111.68		113.43
$(t_3)av$	^{o}F	81.25		83.45
$(t_1)av-(t_3)av$	^{o}F	30.43		29.98
h_1	$B/hr\text{-}ft^2\text{-}^{o}F$	2.95	h_1*	2.91
h_3	$B/hr\text{-}ft^2\text{-}^{o}F$	2.08	h_3*	2.37
h_{av}	$B/hr\text{-}ft^2\text{-}^{o}F$	2.52	h_{av}*	2.64

% increase in h_{av}: 4.76

Table 14. Summary of Driver Input Calibration Data

Test No.	Frequency c.p.s.	q B/hr Before Vibration	Max. q B/hr During Vibration	S B/hr
D-1	137	-0.176	1.813	1.989
D-2	470	0.114	3.080	2.966
D-3	2000	-0.369	2.193	2.562
D-4	6000	-1.562	1.415	2.977
D-5	13100	-0.176	2.299	2.475

Average of Tests No. D-1 through D-5: 2.594 B/hr

Average of Tests No. D-2 through D-5: 2.745 B/hr

Assume constant heat input by driver of 2.75 B/hr over entire frequency range of tests.

Table 15. Thermocouple Calibration Data

Avg. Corrected Standard Thermometer Reading, °F	L&N Conversion Table Rdg., m.v.	Avg. Obsvd. e.m.f., m.v.	Discrepancy
78.04	1.327	1.269	0.058
91.41	1.712	1.636	0.076
101.16	1.998	1.946	0.052
113.64	2.367	2.299	0.068
135.15	3.004	2.921	0.083
156.59	3.639	3.545	0.094
194.92	4.768	4.654	0.114

APPENDIX D

DETERMINATION OF GRASHOF NUMBER

DETERMINATION OF GRASHOF NUMBER

By Beckmann's Method (29):

$$Gr_1 = \frac{\beta g}{\nu^2} D_1^3 (t_1 - t_3).$$

Subscripts have been changed to conform to the notation in Figure 6. Convection is assumed to be negligible when $k_c \cong k$. T is the temperature of the bulk of the gas. Assume $T = \frac{(T_1 + T_3)}{2}$.

The maximum Gr_1 occurs when $(t_1 - t_3)$ is a maximum and when T is a minimum. The maximum $(t_1 - t_3)$ during the experiments is $\cong 40^\circ F$. The minimum T occurs at

$$\frac{(t_1 + t_3)}{2} = \frac{104 + 78}{2} = 91^\circ F.$$

Therefore, assume that $T = 91 + 460 = 550^\circ R$. D_1 is the characteristic length = 1.050 in. $g = 32.2\ (3600)^2\ ft/hr^2$ ν_{air} at $90^\circ F = 0.628$.

$$Gr_1 = \frac{1}{550} \frac{(32.2)\ (3600)^2}{(0.628)^2} \frac{(1.050)^3}{1728} (40) = 51{,}500$$

$$\frac{D_3}{D_1} = \frac{1.494}{1.050} = 1.423$$

From the curve established by Beckmann's experimental results (30), for $D_3/D_1 = 1.423$, $k_c/k = 1$ for Gr_1 up to about 80,000; therefore, convection is negligible at the given conditions.

By Jakob's recommendation (31):

$$A_m = \frac{\pi}{8} H (D_1 + D_3)^2 \text{ for the computation of q.}$$

Using the data for vertical cylinders, as modified by Jakob from Mull and Reiher: $H/L = 16.25/0.222 = 73.2$, where H = height of layer; L = thickness of layer.

$$Gr_L = \frac{gL^3 \beta \Theta}{\nu^2} = 32 (3600)^2 \left(\frac{0.222}{12}\right)^3 \frac{40}{560(0.660)^2} = 431.$$

From the experimentally determined curves of Mull and Reiher (31), $k_c/k = 1$. Therefore convection is negligible.

APPENDIX E

PHYSICAL CHARACTERISTICS OF TEST SECTION

PHYSICAL CHARACTERISTICS OF TEST SECTION

BRASS COMPONENTS

Heating element. $c_p = 0.0928$ B/lb-F (45), $\rho = 546$ lb/ft^3 (46)

Chromalox cartridge heater. 115 v, 800 w.

L: 16 1/4 in. D: 3/4 in.

For heat storage purposes, consider entire heater to be equivalent thermally to a cylindrical shell of brass 1/6 in. thick, OD 0.75 in., ID .417 in.

$$V_h = L\,\frac{\pi}{4}\,(D_1^2 - D_2^2) = \frac{16.25}{12}\,\frac{\pi}{4}\left(\frac{\overline{.75}^2 - \overline{.417}^2}{144}\right) = 0.00284 \text{ ft}^3$$

Tube A. $c_p = c_1 = 0.0928$ B/lb-F, $\rho = 546$ lb/ft^3

3/4 in. Regular Standard Red Brass pipe.

L: 17.75 in. OD=D_1: 1.050 in. ID: 0.822 in.

Thickness: 0.14 in.

$$V_a = L\,\frac{\pi}{4}\,(D_1^2 - D_2^2) = \frac{17.75}{12}\,\frac{\pi}{4}\left(\frac{\overline{1.050}^2 - \overline{.822}^2}{144}\right) = 0.00344 \text{ ft}^3$$

$$A_1 = \pi D_1 L = \pi\left(\frac{1.050}{12}\right)\left(\frac{17.75}{12}\right) = 0.407 \text{ ft}^2$$

Hemispherical plug: 1.050 in. in Dia.

$$V_{pl} = 1/2\left(\frac{\pi D^3}{6}\right) = \frac{\pi}{12}\,\frac{(1.05)^3}{1728} = 0.000175 \text{ ft}^3$$

$$V_1 = V_h + V_a + V_{pl} = 0.00646 \text{ ft}^3$$

$$c_1 \rho V_1 = 0.00646\ (546)\ (0.0928) = 0.32732 \text{ B/F}$$

Tube B. $c_p = c_3 = 0.0931$ B/lb-F; $\rho = 546$ lb/ft^3

1 1/2 in. Extra-Strong Red Brass pipe.

L: 17-11/16 in. OD: 1.900 in. ID=D_3: 1.494 in.

Thickness: 0.203 in.

$$V_b = V_3 = L\,\frac{\pi}{4}\,(D_1^{\,2} - D_2^{\,2}) = \frac{17.69}{12}\,\frac{\pi}{4}\left(\frac{\overline{1.9}^2 - \overline{1.494}^2}{144}\right)$$

$= 0.01096$ ft^3

$$A_3 = \pi D_3 L = \pi\left(\frac{1.494}{12}\right)\left(\frac{17.69}{12}\right) = 0.577 \text{ ft}^2$$

$$c_3\,\rho\,V_3 = 0.0931(546)\ (0.01096) = 0.55713 \text{ B/F}$$

AIR LAYERS

Assume $t_m = 100^\circ$F. $c_p = .24\ \frac{B}{lb\text{-}F} = c_2$

$$\rho = .070\ \frac{lb}{ft^3} = \rho_3$$

Test section annulus

L: 17 in. OD: 1.494 in. ID: 1.050 in.

$$V_t = \frac{17}{12}\ \frac{\pi}{4}\left(\frac{\overline{1.494}^2 - \overline{1.050}^2}{144}\right) = .00873 \text{ ft}^3$$

Driver chamber

L: 1 in. Dia.: 1.494 in.

$$V_d = \frac{1}{12}\ \frac{\pi}{4}\ \frac{(1.494)^2}{144} = .00101 \text{ ft}^3$$

Heater air annulus

$L = 16\ 1/2$ in. OD: .822 in. ID: .75 in.

$$V_x = \frac{16.5}{12}\,\frac{\pi}{4}\left(\frac{\overline{.822}^2 - \overline{.75}^2}{144}\right) = .00085\ ft^3$$

Heater end space:

$L = 7/8$ in. Dia.: .822 in

$$V_e = \frac{7}{8}\,\frac{1}{12}\,\frac{\pi}{4}\,\frac{\overline{.822}^2}{144} = .00027\ ft^3$$

TOTAL AIR SPACE $= V_t + V_d + V_x + V_e = .01086\ ft^3$

$V_2 = V_t + V_d = .00974\ ft^3$

STEEL COMPONENT

Water jacket. 2 1/2 in. Black Steel Pipe.

L: 15 in. OD: 2.875 in. ID: 2.469 in.

Thickness: 0.203 in.

WATER LAYER

OD: 2.469 in. ID: 1.900 in.

$$A = \frac{\pi}{4}\left(\overline{OD}^2 - \overline{ID}^2\right) = .01358\ ft^2$$

Assume water weighs 62.3 lb/ft^3 at $70^\circ F$.

$7.481 \times ft^3 =$ gal.

1 lb. water is raised in temp. $1^\circ F$. by 1 B.

$$\frac{1}{62.33}\,\frac{7.481}{60} = .002 \text{ g.p.m. are raised in temp. } 1^\circ F. \text{ by 1 B.}$$

APPENDIX F

CALIBRATION OF THERMOCOUPLES

CALIBRATION OF THERMOCOUPLES

The nine iron-constantan thermocouples which were used to measure temperatures during subsequent testing were calibrated by comparison with mercury-in-glass thermometers which previously had been certified by the National Bureau of Standards. A controlled-temperature water bath was used to accomplish the calibration.

A large, thermally-insulated container (with a capacity of about five gallons) was used to contain the water bath. The water was stirred continuously by a small, variable-speed electric stirrer. An electric immersion heater, connected to the power supply through a variable output transformer, provided the means for heating the water bath. The nine thermocouple junctions were inserted into a thin-walled glass tube, of small diameter, which was closed at one end. The standard thermometers and the glass tube containing the thermocouple junctions were mounted so that the thermometer bulbs and the closed end of the glass tube were immersed to a depth of several inches into the bath.

During each of seven tests, a constant input to the heater was maintained until there was no appreciable variation of the water temperature. When this condition prevailed, at least two sets of readings were made of the electromotive force in the nine thermocouples and of the temperatures indicated by two standard thermometers. The seven points were selected to represent, at comparatively uniform intervals, the range within which temperatures were expected to fall during subsequent testing (about 75° to 200° F.) The variation between individual thermocouple readings during all tests was within the desired accuracy of about 0.1°F. An average was established of the recorded thermocouple electromotive forces, and of the indicated thermometer readings. The Bureau of Standards error corrections and the corrections recommended by the Bureau of Standards to compensate for the exposed portion of the thermometer stems were applied to each thermometer reading, and an average corrected thermometer reading was determined.

A large graph was constructed, plotting the true temperature as determined from the average of corrected standard thermometer readings, versus the average of recorded thermocouple electromotive forces. The seven points were connected by straight lines. Since the thermocouple wire used was manufactured by Leeds and Northrup Company, data for iron-constantan thermocouple wire, as published by Leeds and Northrup Company, "Standard Conversion Tables," (47) were plotted for comparative purposes at intervals of five degrees, as were also the data for iron-constantan wire as presented by National Bureau of Standards Research Paper 2415, May 1953, (48) plotted at three degree intervals. The curve established from experimental data consistently indicated slightly lower temperatures for a given electromotive force than either of the curves defined by the two published sets of data mentioned above. The average deviation of the experimental curve from the Leeds and Northrup curve was approximately equal to the lower limit of error for acceptability of iron-constantan wire (-66 μv).(49) Average data used in preparing the plot of electromotive force versus temperature in degrees Fahrenheit are listed in Table 15.

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